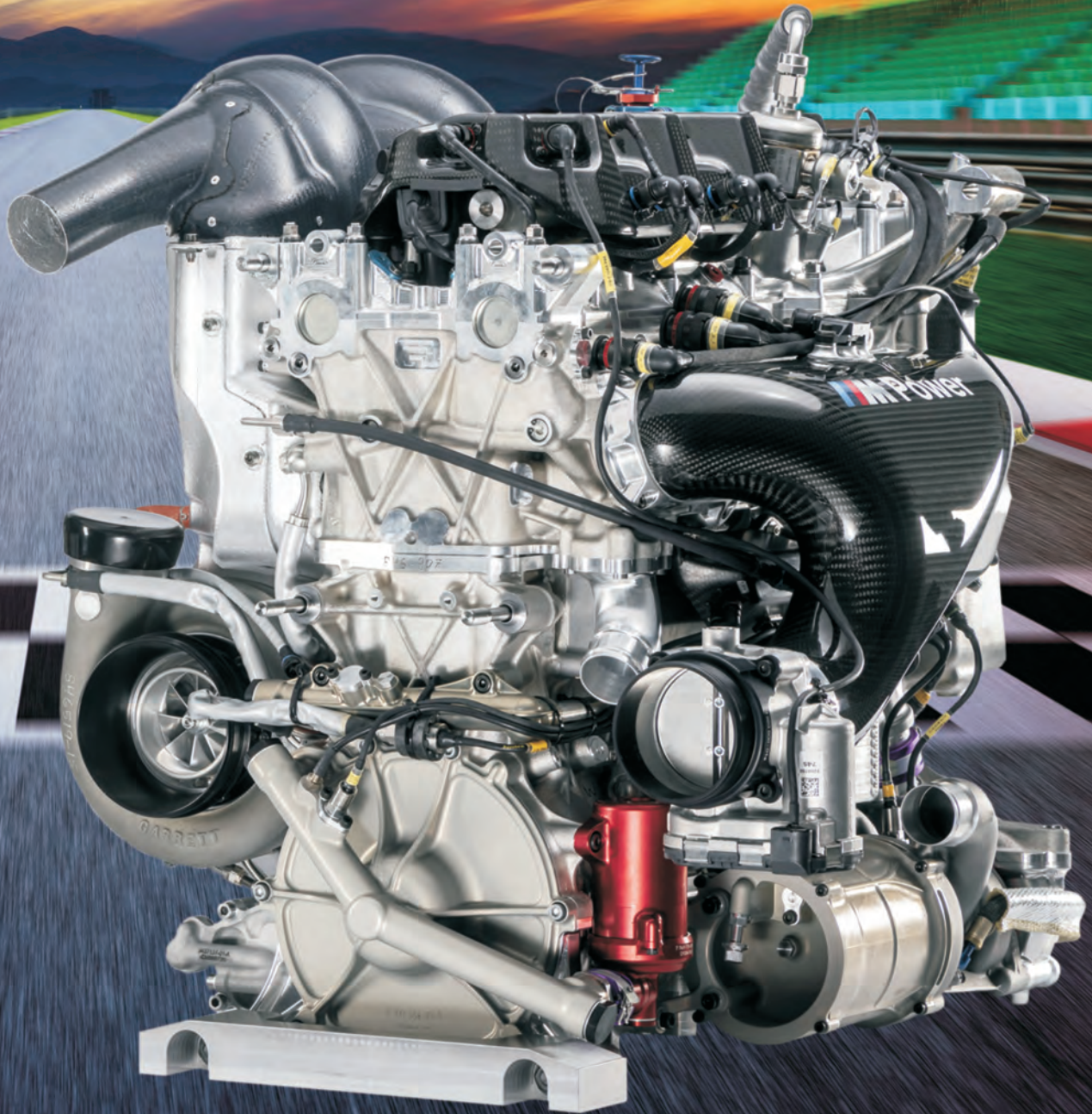




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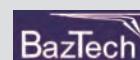
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IV – Multi-Variable Gasoline Engine for Lowest CO₂
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The effect of SiO₂ nanoparticles content in engine oil on tribological properties of valvetrain chain transmission components

The drive from the crankshaft to the camshaft in an internal combustion engine is usually carried out by means of a cogged belt transmission or a chain transmission when high millage is required without service operations. The valvetrains in CI engines sometimes use the gear transmissions, and the historical rather bevel gear can be found in old cars of collectors or in some motorcycle engines. The chain gear used in SI engines has a two- or three-row chain with high strength, due the unevenness of loads that additionally induce chain pulling and valve timing deregulation. The chain transmission requires the use of pre-tensioners, usually self-acting and driven by springs or oil pressure. The vibrations and chain runout are limited using plastic guides placed on the outside of the long straight sections of the chain. The model of the chain transmission developed with the use of the Finite Elements Method, which operates under oil lubrication conditions, was analyzed. Such the model allowed obtaining weight and mass inertial moments of components. The aim of the study was to evaluate the effect of SiO₂ nanoparticles content in engine oil on the friction between chain transmission components. The resulted values of the friction torque in the chain transmission operating in different conditions of lubrication have been presented in the paper.

Key words: valvetrain, chain transmission, wear, friction, SiO₂ nanoparticles

1. Introduction

The combustion engines are commonly used in the most of classic and hybrid vehicles. Their operation is strictly related to the existing valvetrains which can be of cam or camless type. Each cam-type valvetrain contains one or more camshafts. They are driven from the engine crankshaft through a cogged belt transmission or a chain transmission when high millage is required without service operations. The valvetrains in CI engines sometimes use the gear transmissions, and the historical rather bevel gear can be found in old cars of collectors or in some motorcycle engines.

According to [12] chain transmissions are more durable, compact and more efficient than other timing drive systems. If chain dynamics relative to contact stresses, contact forces between various chain drive components exceeds beyond acceptable limits, it leads to vibrations, improper valve and fuel injection timings. Thus, the chain dynamics is the leading source of aggregating Noise, Vibration and Harshness (NVH) issues. If designed valve and injection timings are not achieved, it reduces the volumetric efficiency of the engine and affects overall vehicle performance and fuel economy.

The chain gear used in SI engines contains a one-, two- or three-row chain with high strength, due the unevenness of existing loads during engine operation. Such loads additionally induce chain pulling and can cause valve timing deregulation. The vibrations and chain runout occurring during operation of chain transmission are limited using plastic guides placed on the outside of the long straight sections of the chain. The chain transmission requires the use of pre-tensioners, usually self-acting and driven by springs or drivers using pressure of oil supplied from mean oil line of an engine. Such oil can contain different addi-

tives, i.e. SiO₂ nanoparticles. Components of chain mating with sprockets mounted on camshaft and crankshaft, guides and pretensioner operates under oil lubrication, providing commonly mixture friction conditions. The aim of the study was to evaluate the effect of SiO₂ nanoparticles content in engine oil on the wear and friction of chain transmission components.

2. Tribological aspects of the chain transmission

Tribological aspects of the chain transmission are mainly related to the friction torques and friction forces acted on the transmission components and to wear of them. They depend on the engine loading and speed, lubrication conditions and environment parameters, especially temperature, humidity and pollution existence and possibility to reach the interior of transmission.

According [1], the timing drive and valve train generate approx. 20% of overall engine friction losses depending on the engine speed.

According to [14] the friction caused by chain and guide or sprocket rails is mostly higher than in the case of the toothed-belt timing drive. One of main chain drive disadvantages is that the chain drives reach higher noise level than the toothed belt. It is caused by the impact of the chain links on the sprockets. Comparing the life prediction of the two drive mechanisms, the chain timing drive is the best. Chain damages can occur rarely during engine life and the chain timing drive maintenance is not demanding. Regarding timing drive installations there are no high differences associated with using a toothed belt or chain drive.

Pedersen [17] elaborated the simulation model of the dynamics of roller chain drives using a continuous contact force method. The model of the contact surface between the rollers and sprocket was an important issue regarding the

numerical stability of the simulation program and a model with a real tooth profile proved superior to other applied models. With this model it is possible to perform a dynamic simulation of large marine engine chain drives.

Conwell and Johnson [2] created a new test machine configuration providing more realistic chain loading and allowing link tension and roller-sprocket impact monitoring during normal operation.

Pereira et al. [19] used a multibody methodology to address the kinematic and dynamic effects of roller chain drives was presented. The chain itself was modelled as a collection of rigid bodies, connected to each other by revolute clearance joints. Each clearance revolute joint, representing the connection between pair of links, was made up of the pin link/bushing link plus the bushing link/roller pairs, in case of a roller chain. The problem of contact initialization and its coordination with the numerical integration procedures was treated through controlling the time step size of the numerical integration algorithm in the vicinity of the impact.

Sakaguchi et al. [23] investigated a method for reducing friction loss in the engine timing chain using multi-body dynamics simulation. The method known as the link-by-link model was employed in the simulation to enable representation of the behavior of each single link of the chain and its friction due to contact. A model considering fluctuations in camshaft torque and crankshaft rotational speed was created. This simulation was used to verify the detailed distribution of friction in each part of the chain system as well as the changes of friction in the time domain.

Dwyer-Joyce et al. [5] used a photo elastic stress analysis technique to determine the contact stresses in an automotive chain drive tensioner. The tensioner in normal operation is subject to high magnitude, short duration impact stresses. These stresses are known to cause surface damage, wear, and surface pitting. In order to adequately design the drive system layout, a means for stress quantification is needed. A replica tensioner was made from epoxy resin and tested in a variety of configurations. A simple model was created to relate the chain link load to the resulting tensioner subsurface stress field. This model was used to correlate the observed and predicted location of isochromatic fringes, and hence to evaluate the chain link load from the photo elastic fringe pattern. Once the load was determined the contact model was used to determine the magnitude and location of the resultant peak stresses. Measured impact stresses were several times higher than those calculated from a static analysis.

Weber et al. [26], developed the new test method to facilitate the direct measurement of the real timing chain load under all engine conditions. The measuring principle was based on a strain gauged chain link in combination with a telemetric transducer system. It allowed the continuous observation of the link force during the complete chain revolution in order to identify significant dynamic effects, which are responsible for noise, wear and incorrect valve timing.

Maile et al. [10] used a design for reliability methodology based on the DfSS DCOV process for the development of a cost-effective timing chain drive. The CAE model for

the timing chain drive was used to study the distribution of the chain loads, which provided an essential input both for the concept selection stage and for the development of a reliability model for the timing chain. The DoE study on the CAE model aimed at investigating the significant factors for chain load variability lead to a reliability improvement achieved by reducing the variability in the chain load through revising the tolerances for the sprocket tooth profile. CAE tools and FEA techniques were used for computing chain loads.

Takagishi and Nagakubo [24] showed, that using the longitudinal model the load prediction accuracy was inadequate. Accordingly, a link-by-link model was created, allowing transversal vibration to be considered. As a result, the features of a chain system using a blade tensioner were clarified, thus enabling the chain load and behavior to be predicted with a higher degree of accuracy than before.

Du et al. [4] developed a MBS model of the balancer chain drive. Such a model included a crank train system coupled to a full balancer drive system including the chain drive, balance shafts, and water pump. The nonlinear stiffness and damping effects of the baseline compliant crank chain sprocket and its design iterations were added to the model to understand the basic characteristics and find an optimal setup for the engine start-up period.

3. Lubricants with SiO₂ nanoparticles

Some studies related to the effect of addition of SiO₂ nanoparticles into lubricating oil on its tribological features.

Rashed & Nabhan [21] studied tribological behavior of mineral 20W-50 and semi-synthetic 15W-50 oil dispersing by different amounts (0.5 and 1.0% wt. oil) of SiO₂ nanoparticles. Experiments were carried out on tribometer test-rig under normal load at different temperature values from range 40°C to 100°C. The addition of SiO₂ to engine oils had not reduced the friction coefficient to a great extent.

Li et al. [9] dispersed ultrasonically 0.3% wt. of SiO₂ nanoparticles in ST5W/30 mobile oil. Such a nanolubricant was investigated on a reciprocating tribotester and a four-ball tribotester. As a result, the frictional coefficient decreased in relative to pure ST5W/30 gas mobile oil.

Peng et al. [18] reported, that the SiO₂ nanoparticles oleic dispersed in acid exhibited the similar tribological behavior as diamond nanoparticles in liquid paraffin. During friction the micro-grooves were formed, and then more spherical nanoparticles rolled into the contact area like tiny bearing, balls which reduce the sliding friction.

Nabhan & Rashed [13] studied tribological behavior of two types of lubricant oils dispersing by different amount of SiO₂ and TiO₂ nanoparticles. Experiments were carried out using tribometer test-rig under normal load at different temperatures, 40°C, 80°C and 100°C. The concentrations of Nano oxides additives that used are 0.5% and 1.0% wt. The lowest value of the friction coefficient and wear rate achieved for mineral and semi synthetic lubricant oils with adding 1.0% wt. of TiO₂ at 80°C and 100°C, respectively. The adding SiO₂ to engine oils played only a small role in reducing the friction coefficient.

Prabu et al. [20] studied the tribological characteristics of the commercial SN lubricating oil with addition of SiO₂ nanoparticles. Wear tests were conducted on plain oil and

oil with nanosized SiO₂ additives using a four ball tribometer. It was obtained a considerable reduction in the values of the analyzed: wear scar diameter, wear angle and wear depth for the gear oil added with SiO₂ nanoparticles of 0.4% wt.

According to [3], SiO₂ nanoparticles synthesized by the sol-gel approach were engineered for size and surface properties by grafting hydrophobic chains to prevent their aggregation and facilitate their contact with the phase boundary, thus improving their dispersibility in lubricant base oils. The surface modification was performed by covalent binding of long chain alkyl functionalities using lauric acid and decanoyl chloride to the SiO₂ NP surface. The tribological properties of the dispersions were analyzed under extreme load conditions using a Stanhope Seta Shell Four-Ball E.P. Lubricant Tester working at 1470 rpm shaft speed. The 12.7 mm diameter tests balls with 0.035 μ m roughness were made from AISI 52100 steel. The dendritic structure of the external layer allowed reducing the friction coefficient and wear.

Patil et al. [16] studied the tribological behavior of SiO₂ nanoparticles as additives in Paraffin based SN-500 Base oil. All tests were carried out using pin-on-disk tribotester under variable load and concentration of nanoparticles in lubricating oil. The SiO₂ nanoparticles added into base oil exhibited good friction reduction and anti-wear properties. The SiO₂ nanoparticles in the SN-500 base oil decreased friction coefficient by 61%, 55% and 43% at 0.5% wt. concentration and 36%, 76% and 17% at 0.75% wt. concentration respectively, as compared with standard base oil without SiO₂ nanoparticles. Such tribological behavior was affected by the deposition of nanoparticles on the rubbing surfaces.

4. Modelling of chain transmission

Some models of chain transmissions were elaborated with different level of their complexity.

Novotny and Pišteš [15] stated that computational simulation of chain drives has not been developing too long due to high demands on the computational technology. They focused on simulation of dynamics of the timing chain drive with the use of a multibody system. A mass-produced four cylinder in-line engine with two camshafts and two valves per cylinder has been used as a computational model.

According to [22], a model of roller chain and sprocket dynamics was developed, aimed at analyses of dynamic effects of chain drive systems in automotive valvetrains. Each chain link was modeled as a rigid body with planar motion, with three degrees of freedom and connected to adjacent links by means of a springs and dampers. The kinematics of roller-sprocket contacts were modeled in full detail. Sprocket motions in the chain's plane, resulting from torsional and bending motions of attached camshafts were also considered. One or two-sided guides could be treated as well as stationary, sliding or pivoting tensioners operated mechanically or hydraulically. The model also considered the contact kinematics between chain link rollers and guides or tensioners, allowing for guides/tensioners of arbitrary shape, or simpler (flat and circular) geometries. The model captured the effects of the dynamics of the chain

drive on valvetrain behavior both qualitatively and quantitatively.

According to [7], the timing chain drives simulation results is often represented as families of function graphs (data series). Previously, the analysis of those results was based on static 2D diagrams and animated 3D visualizations. They were suitable for the detailed analysis of a few simulation variants, but not for the comparison of many cases. It was proposed a new approach to the analysis based on coordinated linked views and advanced brushing features. The proposed method introduced a novel, so-called segmented curve view, which could display distributions in families of function graphs. The segmented curve view combined individual function graphs where for a fixed value of the independent variable, a bar extended from minimum to maximum values across the family of function graphs. Each bar was divided into segments (bins) with a color that represents the number of function graphs with the value in that segment.

According to [11] the optimized chain and belt drives can have similar efficiency. While chain drives have higher efficiency than dry belt drives, chain drives and wet belt drives perform similarly. A common dry belt features a strength-to-width ratio of less than half that of a similar chain application. Due to the higher strength-to-width ratio, chain systems featuring a hydraulic tensioner, tight strand guide and slack strand tensioner arm contain all the elements necessary for adapting to additional engine variants with minimal modifications. As a result, design decisions regarding timing drives should be made by considering several different criteria. Adaptability, low dynamic cam oscillation, strength and best-in-class NVH performance can make chain drives the best solution. With proven long-term field durability, chain drives offer compact packaging, optimized efficiency and robustness against dynamic instability.

Egorov et al. [6] proposed the method allowing evaluation of the efficiency of a chain drive with a high measuring frequency rate within a wide range of possible modes of operation as it used the angular acceleration and the acceleration time as controlled variables. The method used the loading rotary bodies attached to a driven shaft of a chain drive to create the rated force in a chain, eliminating a need for a loader. The method allowed determining the mechanical efficiency of chain transmissions without consideration the losses in their bearing units. Determining acceptable levels of the acceleration time of chain drives, allowed evaluation their mechanical efficiency affected by a build quality, lubrication and materials used. Such the efficiency of chain transmissions characterized degradation processes in their elements, and therefore might be a criterion of their lifetime.

Kozlov et al. [8] proposed a new approach to the evaluation of the quality of lubricants for chain transmissions, which was based on the relationship between the chain efficiency pulldown and the use of inappropriate lubricants. To control the energy efficiency of chain drives, it was applied a method using the rotary bodies with the known moments of inertia. The method allowed investigating mechanical losses in a chain drive within a wide range of

speed, load, and transient regimes of operation without the use of strain sensors. It was experimentally proved that the method allows to evaluate the quality of lubricants in the course of operation of a chain transmission.

According to [25], most of the friction in a chain drive comes from the movement of the bushings, pins and rollers. Such friction is a combination of Coulomb friction and viscous damping dependent on the lubrication. It was not included in the model presented in [25], but damping was introduced to obtain a stable solution. As most of the power is transmitted in the tight span, it was assumed that most of the damping also occurs at this side. The damping force is usually much smaller than the stretching force in the chain but is related to this. The damping force in the slack span was neglected, as the stretching force at the slack span was smaller than at the tight span. For simplicity, a viscous damping force was assumed, proportional to the relative velocity between the two end points at the tight side.

The damping force was obtained from the equation (1)

$$F_d = D_{\text{chain}} \cdot [R_{1,p} \cdot d\varphi_p / dt \cdot \cos(\varphi_{1,p} - \theta_{in,p}) - R_{1,g} \cdot d\varphi_g / dt \cdot \cos(\varphi_{1,g} - \theta_{in,g})] \quad (1)$$

The angle φ was used to describe the position of the sprocket (Fig. 1). The rotation coordinates to the first contacting tooth's center-line on the sprocket were φ_p and φ_g for driving and driven sprocket, respectively. To describe the position of a specific i -th roller the angle φ_i was used. The angles between the y axis and the first roller of each sprocket were $\varphi_{1,p}$ and $\varphi_{1,g}$ – for driving and driven sprocket, respectively. The angles $\theta_{in,p}$ and $\theta_{in,g}$ were the chain angles, relative to the x axis, of the first links in contact with the driving and driven sprocket, respectively.

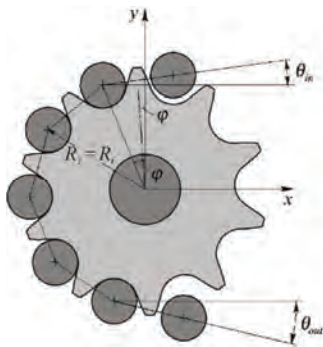


Fig. 1. Sprocket in an arbitrary load position

5. Materials and methods

5.1. Model of analyzed chain transmission

The analyzed chain transmission was one from the SI combustion engine Toyota Corolla 1.4 VVTI (Fig. 2). In the present study the model contained three assemblies:

- the assembly 1 of the crankshaft with its sprocket and its flywheel,
- the assembly 2 of the outlet camshaft with its sprocket,
- the assembly 3 of the inlet camshaft with the phaser and its sprocket, was elaborated with the Finite Element Method. The mentioned assemblies were connected through the one row chain 4.



Fig. 2. The chain transmission components

The camshaft assemblies, crankshaft assembly were treated as stiff solid bodies connected with bearings through contact elements. The crankshaft assembly rotated with constant rotational speed oftenest equal 120 rpm but for one case the full chain assembly rotated also with such a speed equal 60 rpm. The chain was modelled as the set of solid rigid components connected by contact elements. The model components allowed determining their weights and mass inertia moments.

5.2. Experimental stand for friction studies in the chain transmission

The friction in the chain transmission was determined using the stand utilized the original SI engine with some modifications made. The crankshaft assembly was bearing in two border mean journal bearings of crankshaft, in which the original bronze half-shells were replaced by ones made of PTFE. This was to reduce the friction coefficient between the crankshaft journals and their half-shells.

The outlet and inlet camshaft assemblies were bearing in their border pairs of journal bearings. To decrease the friction coefficient, in the bushing covers of such bearings the threaded holes were made to place grease nipples supplying lithium grease into contact zones between camshaft journals and their bushings. The grease was continuously replenished in the bearings using a grease gun.

The self-starter driving flywheel connected with camshaft was placed in the engine body through the set of screws, which was a little different than in the original vehicle. The self-starter was supplied with voltage from the battery. The chain is tensioned in real by the tensioner piston supplied by the oil pressure and acting on swingable guide. However, in the stand the piston with top plane is replaced by the piston with a roller. The position of the latter can be changed manually, what allowing quick disconnection of the front piston plane from the pressing guide.

The crankshaft and camshaft subassemblies were consecutively dismantled from the engine for replacing the sprocket on the pulley or rearranging planed components in the given assembly and re-mounted in the engine for further testing.

5.3. The determination of total friction torque in the chain transmission

The total friction torque in chain transmission was determined in several steps.

First, the crankshaft subassembly was arranged to be with the flywheel and with sprocket. Its mass inertia moment was equal I_1 . The crankshaft subassembly was driven by self-starter to the predetermined rotational speed of 120 rpm and then braked by the friction torque M_{T1} occurring in two crankshaft main bearing up to stop. The time t_1 of braking that assembly was obtained and registered.

Next, the crankshaft sprocket and the outlet camshaft subassembly sprocket were replaced with the pullies. Such pullies were connected by the belt tensioned by the roller of the tensioner piston. The crankshaft assembly was driven by the self-starter to the rotational speed of 120 rpm and, via the belt tensioned by the tensioner roller, the outlet camshaft subassembly was also driven. The mass inertia moment of the outlet camshaft assembly with its pulley was equal to $I_2 + I_{20}$. The contact pressure between the roller and the belt was removed via manual displacement of the roller relative to the belt and as a result the outlet camshaft subassembly was braked by the friction torque M_{T20} occurring in its two bearing up to stop. The time t_2 of braking that assembly was obtained and registered.

Then the procedure was repeated for the outlet camshaft subassembly containing its pulley connected with its sprocket through additional screws. The mass inertia moment of the outlet camshaft subassembly containing its pulley and sprocket was equal to $I_2 + I_{20} + I_{21}$. The time t_3 of braking that assembly by the friction torque M_{T201} occurring in its two bearing up to stop was obtained and registered.

Next, the procedure was repeated for the inlet camshaft assembly first containing the phaser and the sprocket connected with the pulley based on the phaser outer cylindrical surface. The mass inertia moment was equal $I_3 + I_{30}$. The time t_4 of braking that assembly by the friction torque M_{T30} occurring in its two bearing up to stop was obtained and registered. In the following step, the procedure was repeated for the inlet camshaft assembly containing the phaser, and the sprocket connected with the pulley, which was based on the phaser outer cylindrical surface and connected with the additional heavy ring through additional screws. The mass inertia moment was equal $I_3 + I_{30} + I_{31}$. The time t_5 of braking that assembly by the friction torque M_{T301} occurring in its two bearing up to stop was obtained and registered.

Then the angular decelerations $\varepsilon_1, \varepsilon_2, \varepsilon_3, \varepsilon_4, \varepsilon_5$ were obtained from the equations (2)–(6) and the inertia moments were determined from the equations (7)–(11).

$$\varepsilon_1 = \omega_0 / t_1 \quad (2)$$

$$\varepsilon_2 = \omega_0 / t_2 \quad (3)$$

$$\varepsilon_3 = \omega_0 / t_3 \quad (4)$$

$$\varepsilon_4 = \omega_0 / t_4 \quad (5)$$

$$\varepsilon_5 = \omega_0 / t_5 \quad (6)$$

$$M_{T1} = I_1 \cdot \varepsilon_1 \quad (7)$$

$$M_{T20} = (I_2 + I_{20}) \cdot \varepsilon_2 \quad (8)$$

$$M_{T201} = (I_2 + I_{20} + I_{21}) \cdot \varepsilon_3 \quad (9)$$

$$M_{T30} = (I_3 + I_{30}) \cdot \varepsilon_4 \quad (10)$$

$$M_{T301} = (I_3 + I_{30} + I_{31}) \cdot \varepsilon_5 \quad (11)$$

Finally, the self-starter drove up to the predetermined crankshaft rotational speed the full assembly containing subassembly of crankshaft with flywheel and sprocket and inlet camshaft subassembly with original phaser and sprocket and the outlet camshaft subassembly with the original sprocket connected through original chain mating with its guides and tensioned by the modified tensioner piston. Then self-starter was turned off and the full assembly was braked up to stop by the friction torques: M_{T1} in two crankshaft bearings, M_{T21} in two outlet camshaft bearings, M_{T3} in two inlet camshaft bearings, respectively and the M_{int1} , M_{int2} and M_{int3} friction torques occurred in three groups of segments of the chain transmissions (Fig. 3). The time t_6 of braking the chain assembly was obtained and registered. The angular deceleration ε_6 was obtained from the equation (12).

$$\varepsilon_6 = \omega_0 / t_6 \quad (12)$$

It was assumed for each case of calculations, that the friction coefficients μ_a and μ_b in the border bearings ‘a’ and ‘b’ were constant.

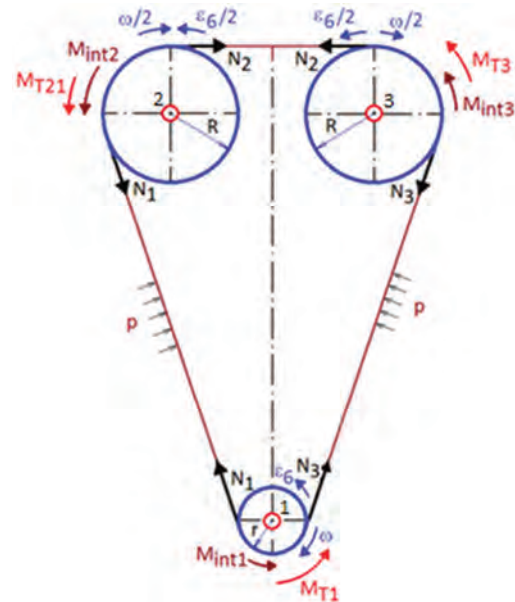


Fig. 3. The model of loading the chain transmission components

The reactions R_{a3s} , R_{b3s} and the friction torque M_{T3} were determined using the equations (13)–(15) and the model of loading the bearings in the assembly, the total weight of which was a sum of the inlet camshaft weight G_3 and the weight G_{3s} of the phaser – sprocket subassembly (Fig. 4a). Additionally, the reactions R_{a30} , R_{b30} and the friction torque M_{T30} were determined using the equations (16)–(18) and the model of loading the border bearings in the assembly, the total weight of which was a sum of the inlet camshaft weight G_3 , the weight G_{3s} of the phaser – sprocket subassembly and the weight G_{30} of the pulley embedded on the outer cylinder surface of the phaser (Fig. 4b). Also the reactions R_{a301} , R_{b301} and the friction torque M_{T301} were determined using the equations (19)–(21) and the model of loading the border bearings in the assembly, the total weight of which was sum of the inlet camshaft weight G_3 ,

the weight G_{3s} of the phaser – sprocket subassembly, the weight G_{30} of the pulley and the weight G_{31} of the heavy ring connected to the pulley (Fig. 4c). From equations (10)–(11) and (16)–(21), the friction coefficients μ_a and μ_b were determined. Then, the friction torque M_{T3} was obtained from equations (13)–(15).

The reactions R_{a21} , R_{b21} and the friction torque M_{T21} were determined using the equations (22)–(24) and the model of loading the bearings in the assembly, the total weight of which was a sum of the outlet camshaft weight G_2 and the weight G_{21} of the sprocket (Fig. 4d).

$$R_{a3s} + R_{b3s} = G_3 + G_{3s} \quad (13)$$

$$R_{a3s} \cdot (a + b) - G_3 \cdot b + G_{3s} \cdot c = 0 \quad (14)$$

$$M_{T3} = R_{a3s} \cdot \mu_a \cdot R_B + R_{b3s} \cdot \mu_b \cdot R_B \quad (15)$$

$$R_{a30} + R_{b30} = G_3 + G_{30} \quad (16)$$

$$R_{a30} \cdot (a + b) - G_3 \cdot b + G_{30} \cdot c = 0 \quad (17)$$

$$M_{T30} = R_{a30} \cdot \mu_a \cdot R_B + R_{b30} \cdot \mu_b \cdot R_B \quad (18)$$

$$R_{a301} + R_{b301} = G_3 + G_{30} + G_{31} \quad (19)$$

$$R_{a301} \cdot (a + b) - G_3 \cdot b + (G_{30} + G_{31}) \cdot c = 0 \quad (20)$$

$$M_{T301} = R_{a301} \cdot \mu_a \cdot R_B + R_{b301} \cdot \mu_b \cdot R_B \quad (21)$$

$$R_{a21} + R_{b21} = G_2 + G_{21} \quad (22)$$

$$R_{a21} \cdot (a + b) - G_2 \cdot b + G_{21} \cdot c = 0 \quad (23)$$

$$M_{T21} = R_{a21} \cdot \mu_a \cdot R_B + R_{b21} \cdot \mu_b \cdot R_B \quad (24)$$

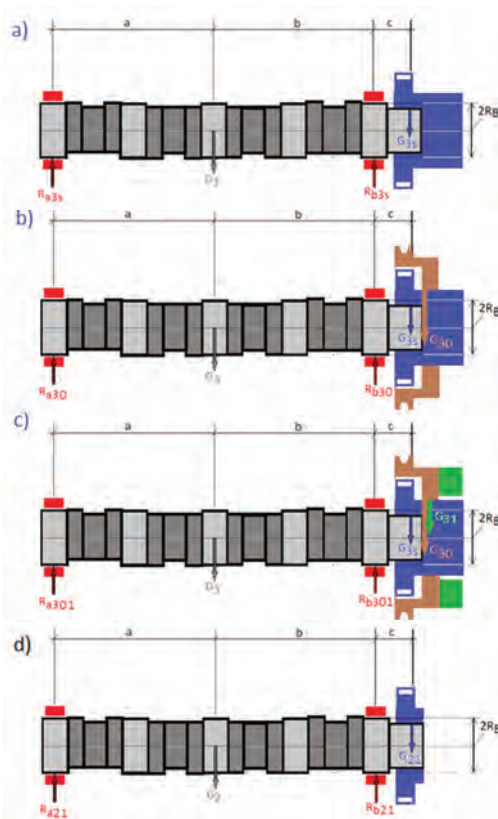


Fig. 4. Model for loading the bearing in the assembly containing: a) inlet camshaft, phaser and sprocket, b) inlet camshaft, phaser, sprocket and pulley, c) inlet camshaft, phaser, sprocket, pulley and heavy ring, d) outlet camshaft and sprocket

The internal friction torque M_{int} in the chain transmission was estimated using the scheme in the Figure 1 and the equations (25)–(36). It was assumed, that such torque contains three components: M_{int1} – related to the chain resistance to motion occurring in the crankshaft sprocket vicinity, M_{int2} – related to the chain resistance to motion occurring in the outlet camshaft sprocket vicinity, and M_{int3} – related to the chain resistance to motion occurring in the inlet camshaft sprocket vicinity.

It was also assumed, that such components are equal M_{int0} . The friction torques in the crankshaft bearing M_{T1c} , in the outlet camshafts bearing M_{T21c} and in the inlet camshaft bearing M_{T3c} during breaking of the studied full chain transmission assembly can be estimated from the equations (28). In the equation (29), r was the pitch radius for the crankshaft sprocket and R was the pitch radius for the inlet camshaft sprocket and for the outlet camshaft sprocket.

$$(N_1 - N_3) \cdot r = M_{T1c} + I_1 \cdot \epsilon_6 + M_{int1} \quad (25)$$

$$(N_2 - N_1) \cdot R = M_{T21c} + (I_2 + I_{21}) \cdot (\epsilon_6 / 2) + M_{int2} \quad (26)$$

$$(N_3 - N_2) \cdot R = M_{T3c} + I_3 \cdot (\epsilon_6 / 2) + M_{int3} \quad (27)$$

$$M_{T1c} = M_{T1}, M_{T21c} = M_{T21}, M_{T3c} = M_{T3} \quad (28)$$

$$M_{T1} \cdot R / r + I_1 \cdot \epsilon_6 \cdot R / r + M_{int1} \cdot R / r + M_{T21} + (I_2 + I_{21}) \cdot (\epsilon_6 / 2) + M_{int2} + M_{T3} + I_3 \cdot (\epsilon_6 / 2) + M_{int3} = 0 \quad (29)$$

$$R / r = 2 \quad (30)$$

$$M_{int} = M_{int1} + M_{int2} + M_{int3} \quad (31)$$

$$M_{int1} = M_{int2} = M_{int3} = M_{int0} \quad (32)$$

$$M_{int} = 3 \cdot M_{int0} \quad (33)$$

$$2 \cdot M_{T1c} + 2 \cdot I_1 \cdot \epsilon_6 + 2 \cdot M_{int1} + M_{T21c} + (I_2 + I_{21}) \cdot (\epsilon_6 / 2) + M_{int2} + M_{T3c} + I_3 \cdot (\epsilon_6 / 2) + M_{int3} = 0 \quad (34)$$

$$2 \cdot M_{T1c} + 2 \cdot I_1 \cdot \epsilon_6 + (4 / 3) \cdot M_{int} + M_{T21c} + (I_2 + I_{21}) \cdot (\epsilon_6 / 2) + M_{T3c} + I_3 \cdot (\epsilon_6 / 2) = 0 \quad (35)$$

$$M_{int} = - (3 / 4) \cdot [2 \cdot M_{T1c} + M_{T21c} + M_{T3c} + 2 \cdot I_1 \cdot \epsilon_6 + (I_2 + I_{21} + I_3) \cdot (\epsilon_6 / 2)] = 0 \quad (36)$$

5.4. The determination of the friction torque between chain components

The friction torque occurring between chain roller and pin was estimated experimentally. The friction between of neighboring chain segments (outer one and inner one) in each such a pair was neglected. It was assumed, that two rollers with two inner segments create one rigid body and two pins with two outer segments create the other rigid body. The friction torque was measured using the concept of the physical pendulum.

Before measurement the chain was placed in the ultrasonic bath for the 3 min to remove the residual particles of lubricant existing in the contact zones between rollers and pins of the chain.

Then one chain pin is fixed, and the chain segments swing relative to the axis of the fixed roller. To make set of chain segments close to one rigid body, with mass inertia moment I_{ch} and weight m_{ch} , the chain segments were wrapped by several layers of foil, weight of which was neglected.

The tilts of the chain from the initial vertical position were below $\alpha = 30$ deg to meet the conditions of small deviations. After tilting of the chain from its initial position by $\alpha_0 = 30$ deg, it was measured the time and number n of swingings of the pendulum up to the rest.

Then the pin was unfixed and it with the mating roller was immersed in the engine oil to introduce lubricant into contact zone between the pin and the roller and then the pin was re-fixed. Next, the measurement of the pendulum swingings was repeated.

Then the pin was once again unfixed and placed consecutively: in the kerosene, the methanol and finally in the ultrasonic bath to remove the residual particles of the pure engine oil. Next, the roller together with the pin was immersed into the engine oil with 6% wt. amount of SiO₂ nanoparticles. After re-fixing of the pin, the measurement of the pendulum swingings was repeated.

The motion of the physical pendulum was described by the equation

$$I_{ch} \cdot d^2\alpha / dt^2 + \mu \cdot r_r \cdot \text{sign}(\alpha) \cdot [m_{ch} \cdot g \cdot \cos\alpha - m_{ch} \cdot r_r \cdot (d\alpha / dt)^2] = m_{ch} \cdot g \cdot l \cdot \sin\alpha \quad (37)$$

where: l – distance between the axis of fixed pin and the center of gravity of chain arranged to one rigid body, r_r – radius of chain pin, t – time, μ – friction coefficient in the contact zone between chain roller and pin.

Integration of the equation (37) allowed estimation of friction torque in the contact between the chain pin and mating roller from the equation (38)

$$M_R = \alpha_0 \cdot m_{ch} \cdot g \cdot l / (2 \cdot n) \quad (38)$$

The angular frequency of physical pendulum swinging was estimated from the equation (39).

$$\omega = 2 \cdot \pi \cdot n / t_0 = (m_{ch} \cdot g \cdot l / I_{ch})^{0.5} \quad (39)$$

According to the equation (1), the friction between chain components is related to the speed of their motion. The internal friction torque M_{int} in the chain transmission is also related to the friction torque M_r between chain rollers and pins. During operation the estimated number k of chain rolls clearly sliding against mating pins was obtained from the equation (40).

$$k = \pi \cdot (r + 2 \cdot R) / p \quad (40)$$

where: p – chain pitch.

It was assumed, that for the constant value of rotational speed ω_1 of the crankshaft, the dry internal torque M_{int} (M_{rdry} , ω_1) in the chain transmission is related to the dry friction torque M_{rdry} between chain roller and pin obtained using physical pendulum method according to the equation (41).

$$M_{int}(M_{rdry}, \omega_1) = A \cdot k \cdot M_{rdry} + B \cdot \omega_1 \quad (41)$$

For the same value of speed v , the relation between internal friction torque $M_{int}(M_{roil}, v)$ in the chain transmission lubricated by the engine oil 10W-40 and the friction torque M_{roil} in contact zone between chain roller and pin lubricated with the same oil obtained using physical pendulum method can be estimated from the equation (42).

$$M_{int}(M_{roil}, \omega_1) = A \cdot k \cdot M_{roil} + B \cdot \omega_1 \quad (42)$$

For the same value of speed v , the relation between internal friction torque $M_{int}(M_{rsio2}, v)$ in the chain transmission lubricated by the engine oil 10W-40 with 2% wt. amount of SiO₂ nanoparticles is related to the friction torque M_{rsio2} in contact zone between chain roller and pin lubricated with the same mixture obtained using the physical pendulum method can be estimated from the equation (43):

$$M_{int}(M_{rsio2}, \omega_1) = A \cdot k \cdot M_{rsio2} + B \cdot \omega_1 \quad (43)$$

For two values of crankshaft rotational speed, namely $\omega_1 = 0.12$ rad/s and $\omega_1 = 0.06$ rad/s, two different values of the dry internal torque $M_{int}(M_{rdry}, \omega_1)$ were determined from the equation (41) and then the constants A and B were estimated.

6. Results and discussions

The values of weights and mass inertia moment were presented in the Table 1. The highest weight and the mass inertia moment had the crankshaft. The weight and mass inertia moment for the inlet camshaft were higher than for the inlet one, as the inlet camshaft assembly contained the heavy phaser.

Table 1. Weights and mass inertia moments of chain transmission components

Component	Weight M	Mass inertia moment Iz against rotation axis z
–	kg	kgm ²
crankshaft	11.887	0.137593
crankshaft sprocket	0.047	0.000007
outlet camshaft	2.379	0.000341
outlet sprocket	0.165	0.000168
outlet pulley	0.172	0.000188
inlet camshaft	3.822	0.001628
inlet pulley	0.733	0.000839
inlet ring	0.858	0.000993
chain	2.496	0.002049

The values of friction torque in the bearings were presented in the Table 2. The obtained values of internal dry friction torque in the chain transmission were of 26% lower than the values of friction torque in the two modified crankshaft bearings containing PTFE shells.

Table 2. Calculated values of friction torque in bearings and internal friction torque in the chain transmission

Friction torque	Values
–	Nm
M_{T1}	0.288174
M_{T20}	0.003196
M_{T201}	0.002835
M_{T3}	0.004473
M_{T30}	0.003758
M_{T301}	0.004466
$M_{int}(\omega_1 = 0.12 \text{ rad/s})$	0.169358
$M_{int}(\omega_1 = 0.06 \text{ rad/s})$	0.167130

The resulted courses of the swinging angle α of the chain arranged as a physical pendulum against time t for different cases of friction occurring in the contact zone between fixed chain pin and mating chain roller, namely: dry, lubricated

with pure engine oil and with engine oil containing 2 wt.% of SiO₂ nanoparticles was presented in the Fig. 5.

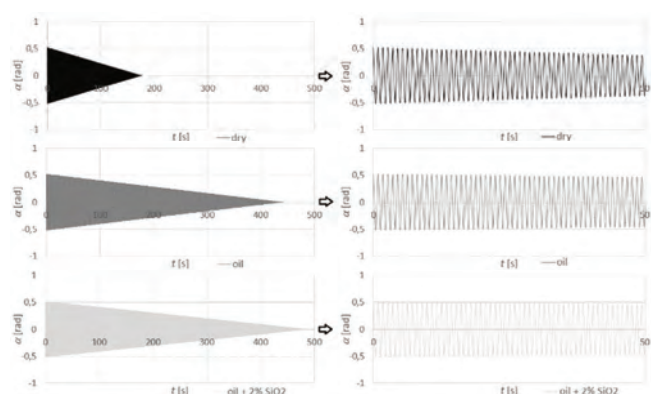


Fig. 5. The swinging angle α of the chain arranged as a physical pendulum against time t for different cases of friction in the contact zone between fixed chain pin and mating chain roller: dry, lubricated with pure engine oil and with engine oil containing 2wt % of SiO₂ nanoparticles. Courses on the right are the increased views for courses on the left side in the figure

Table 3 presents the values of friction torque obtained from the pendulum swingings for different friction in the contact zone between fixed chain pin and chain roller mating with it. The friction torque values for the in the contact zone lubricated by the engine oil were 2.5 times lower than for the dry friction. The addition of 2% wt. amount of SiO₂ nanoparticles into engine oil 10W-40 resulted in the decrease of friction torque about 10%. The constants A and B were estimated to be equal 14 and 1.22 s/m, respectively. Table 3 also included the estimated values of the internal friction torque in the chain transmissions. The value of

internal friction torque in the chain transmission lubricated by the engine oil was lower by 58% than in case of operation under the dry friction conditions. The addition of 2 wt.% amount of the SiO₂ nanoparticles into the engine oil decreased internal friction torque by 9% in comparison to the lubrication by pure engine oil.

Table 3. Friction torque for different friction in the contact zone

Operational conditions in contact zone	Friction torque in the contact zone between chain pin and roller	Internal friction torque in chain transmission
	Nm	Nm
Dry	0.00025	0.169326
Lubricated with engine oil	0.00010	0.070365
Lubricated with engine oil containing 2 wt.% amount of SiO ₂ nanoparticles	0.00009	0.063768

7. Summary

The measurement using physical pendulum swingings method is the useful method for the estimation of friction torque in the contact zone between chain pin and roller for different lubrication conditions. The internal friction torque in the chain transmission lubricated by the engine oil can be smaller about 58% than in the case of lack of engine oil. Addition of 2% wt. amount of the SiO₂ nanoparticles into the engine oil decreased internal friction torque by the 9% in comparison to the case of engine oil without such SiO₂ nanoparticles.

Nomenclature

AISI American Iron and Steel Institute
CAE computer aided engineering
DfSS design for Six Sigma
DoE design of experiment
DCOV define – characterize – optimize – verify
FEA finite element analysis
MBS multibody dynamics system

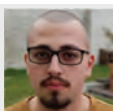
NVH noise, vibration and harshness
NP nano-particles
PTFE poli(tetrafluoroetylen)
SI spark ignition
SN synthesised
VVTI variable valve timing with intelligence

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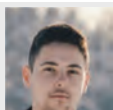
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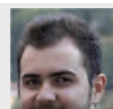
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Experimental research and CFD analysis of flow parameters in a SCR system for the original part and WALKER's replacement

The article presents the results of experimental research and their comparison with CFD simulations for the original selective catalytic reduction system and WALKER replacement. The research was performed to develop the WALKER universal mixer. The SCR prototype without mixer and with the proposed mixer were tested and compared with the original VW part. The next step was reverse engineering, which consisted in scanning the tested parts with a laser and processing their point cloud in Leios2 program. Reverse engineering has allowed the reconstruction of 3D geometry of the tested parts in the Catia v5 program and then preparation their models for computational fluid dynamics. Numerical simulations were carried out in the Ansys Fluent program, thanks to which several quantities were determined e.g. uniformity index of gas flow through the monolith and coefficient of variation as a measure of mixing degree, which have a significant impact on the design of the mixer and the SCR system.

Key words: selective catalytic reduction, computational fluid dynamic, static mixer, uniformity index, coefficient of variation

1. Introduction

At the present day, a future without diesel engines does not seem real for light, heavy duty vehicles and off-road machinery. For such a large number of applications, there are no alternatives when it comes to environmental and economic objectives. Acceptance of this trend by society and customers, causes tightening of euro standards (Table 1), mainly by reducing NO_x , PM and use of other test methods but also entering a limit for CO_2 emissions (95 g/km until 2021) [6]. Therefore, the diesel engines are constantly improving as well as the exhaust system in terms of fuel consumption and reducing of harmful exhaust gas components.

Table 1. Euro 5 and 6 standards for a diesel engine – new types approval for M vehicle category [6, 10]

Euro standard	Date from	Test cycle	NO_x [g/km]
5a	9.09	NEDC	0.18
5b	9.11	NEDC	0.18 **
6b	9.14	NEDC	0.08
6c	—*	WLTP	0.08
6d-temp	9.17	WLTP + RDE	CF = 2.1:WLTP 0.08;RDE 0.168
6d	1.20	WLTP + RDE	CF = 1.5:WLTP 0.08;RDE 0.120

* For new vehicles – 1st registration from 09.2018
**Changed PM from 5.0 to 4.5 mg/km and added PN 6.0*E11 #/km

From Euro 5a to 6b standard, diesel cars had significant divergence between the results of the NEDC (New European Driving Cycle) laboratory test and reality. For this reason, the Worldwide Harmonized Light Vehicle Test Procedure (WLTP) has been introduced which refers to the actual conditions in which drivers travel. The complement of this laboratory test is the Real Driving Emission (RDE) which is road test using Portable Emission Measurement System (PEMS). For the automotive industry to be allowed sufficient lead-time to adapt its strategies for new test cycle, a Conformity Factor (CF) has been introduced which is the maximum ratio between the emission limit in the laboratory test and measured in the road test [6, 10].

To fulfill this rising challenge, there are two main technologies to reduce the NO_x from diesel engine gases [1].

The first of them is the Lean NO_x Traps (LNT) or NO_x Storage Catalyst (NSC). Reactors store NO_x under lean condition, then under the fuel-rich engine condition, the NO_x are released and reduced from base of barium carbonate (BaCO_3). LNT technologies are insufficient to provide desired NO_x emission reduction, especially for engines with larger capacities [3, 11].

The second most used technology for reduction of nitrogen oxides over 90% from exhaust gases of diesel engine is the SCR (Selective Catalytic Reduction) system (Fig. 1). This method uses an injector to inject the urea-water solution (UWS) also known as AdBlue or Diesel Exhaust Fluid (DEF) into the hot exhaust upstream of the SCR monolith. Due to limited amount of space, more and more often meets ones called the SDPF (SCR-Catalyzed Diesel Particulate Filter), which means the SCR coated onto a DPF [11, 12].

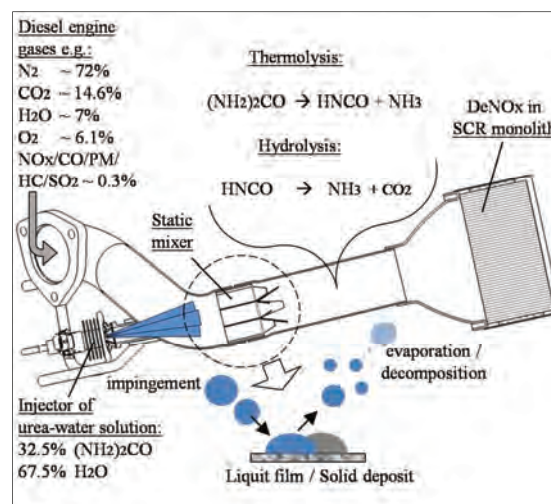


Fig. 1. Sketch showing the physical and chemical processes in SCR system

The UWS are heated up by exhaust gas and decomposes (thermolysis) to ammonia (NH_3) and isocyanic acid (HNCO), which in turn reacts (hydrolysis) with water vapour (H_2O) to ammonia and carbon dioxide (CO_2) [1, 15].

At the low exhaust gas temperature (before light-off) on the supercooled surfaces, tip of injector, mixer or inlet surface of monolith a liquid layer (wall film) of UWS crystallizes after evaporation of water, forming deposit (e.g. biuret, cyanuric acid). This may lead to reducing catalyst activity or increasing engine backpressure [3, 9].

The most important NO_x reactions are called standard, fast and slow and their priority depends on NO_2/NO_x ratio (increasing DeNO_x up to 0.5 ratio). They mainly turn into to harmless nitrogen (N_2) and water [7]. The required amount of UWS dosing for complete conversion of NO_x can be determined by means of the feed ratio α defined as:

$$\alpha = \frac{\text{NH}_{3,\text{in}}}{\text{NO}_{x,\text{in}}} \quad (1)$$

where: $\text{NH}_{3,\text{in}}$ – amount of NH_3 molecules from the urea, $\text{NO}_{x,\text{in}}$ – amount of NO_x molecules in exhaust gases.

The stoichiometric feed ratio of $\alpha = 1$ is theoretically required for complete conversion of NO_x [11, 15].

Good SCR systems combine high NO_x conversion with low ammonia slip, which is undesired emission. To meet these requirements, there are three main types of SCR reactors, which differ in some properties. Vanadia SCR catalysts contain V_2O_5 (1–3%) as the active component, typically impregnated on an anatase TiO_2 support together with WO_3 to stabilize the vanadia and increase thermal durability. $\text{V}_2\text{O}_5\text{-WO}_3/\text{TiO}_2$ characterizes high conversion in medium range temperatures (250–550°C). Cu-zeolite SCR catalysts (Cu-ZSM-5), exhibit higher NO_x conversion at low temperatures (200–650°C), while Fe-zeolite SCR catalysts (Fe-ZSM-5), show higher NO_x reduction at high temperatures (350–700°C). Furthermore, Cu-ZSM-5 catalysts have the greatest possibility of NH_3 storage [7, 11].

The best way to achieve the optimal NO_x aftertreatment system is presented in Fig. 2. The most important part of the development process flow is the initial engineering. It consists of the definition of goals (emission regulation, type of UWS, level of NO_x reduction and backpressure), quantifying inputs (engine emission data, allowable packaging space and work cycle) and development of the initial design. The next step is simulation that allows quickly evaluate designs without the additional costs associated with building hardware and running tests. After this step, hardware testing should be performed to evaluate NO_x reduction and deposit formation. Following a defined process will help to complete NO_x aftertreatment system, meeting the emission certification and saving time and money [4].

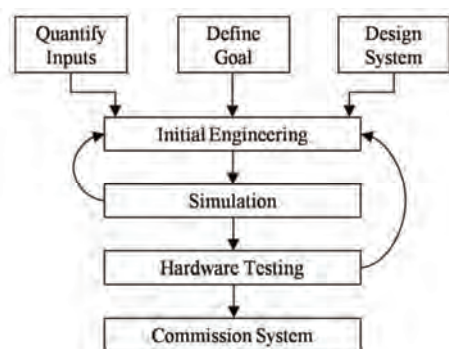


Fig. 2. SCR development process flow

Focusing attention on the process of designing a mixer (Fig. 3), many parameters affect the performance of a mixer such as tube diameter, injector types, injector mounting angle, mass flow rate, temperature, mixing length etc. As mentioned in the study of Zheng et al. [15], it is unrealistic to expect a mixer to perform well in all applications without carefully tuning it to a given exhaust environment. Therefore, development cycle starts with new 3D concept through CFD then prototypes and laboratory tests are initiated to confirm analytical results. Finally, deposit checking and durability are studied and eventually changes are introduced [15].

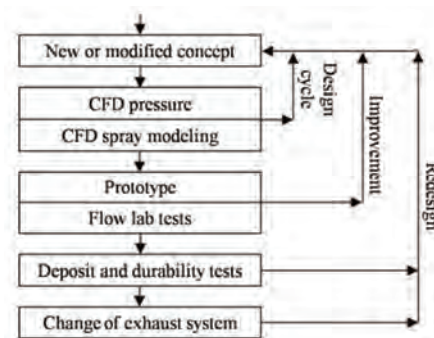


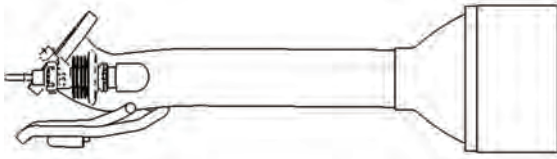
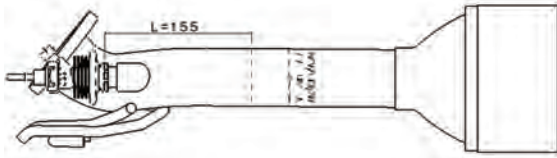
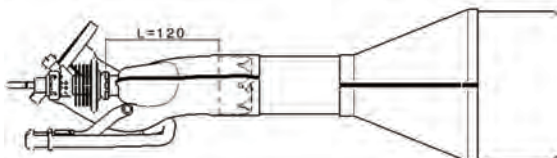
Fig. 3. Mixer development process flow

2. Experimental

The aim of the research within this study was to compare the WALKER SCR replacement prototype for after-market (AM) with the original SCR part (for VW Passat 2.0 BlueTDi 103kW, 2010-2014, fulfilling Euro 6, common rail) [2]. The SCR system consisted of two SCR reactors, however in this paper we focused on the first of them (front part with an injector and a mixer).

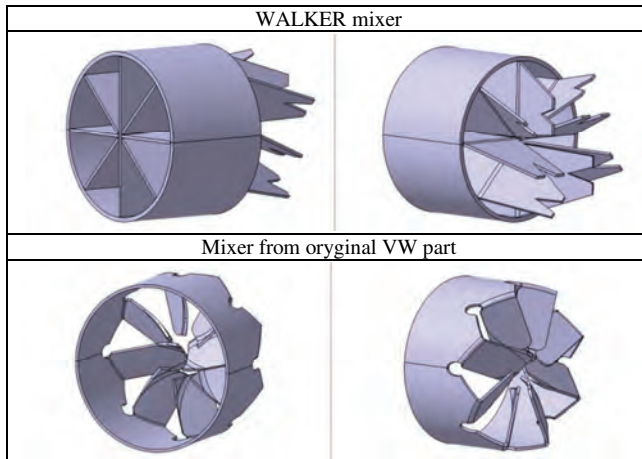
Several experimental studies and all numerical simulations have been performed according to cases, as presented in Table 2. The SCR system was examined without mixer and with proposed WALKER mixer. What's more, they were compared with the original VW part and mixer.

Table 2. Investigated cases

Case 1: WALKER prototype without mixer

Case 2: WALKER prototype with mixer; injector-mixer distance $L = 155$

Case 3: Original VW part with injector-mixer distance $L = 120$


Tested static mixers (Table 3) were prepared in the Catia v5 program. Concept of WALKER mixer was planned for ease of assembly and universal operation. The mixer of VW original product was prepared on the basis of a reverse engineering, which will be discussed in the next section.

Table 3. Tested static mixers



The tested prototype had a vanadium SCR catalyst (V_2O_5) as the active component impregnated and supported on an anatase and silica (TiO_2-SiO_2). The tested monolith had dimensions: $5.66 \times 3''$ (diameter $\times$ length [inch]) with cell density 300 CPSI (cell/inch²) and 9 mil (0.23 mm) wall thickness.

The pressure drops in the prototype and its mixer were measured in a laboratory test facility. The test stand (Fig. 4), consisted of feed tube with mass flowmeter of air inflow (in eight growing values) and micro manometer (for static pressure) installed at the inlet side. Furthermore, the test stand allows also for measuring the temperature of air at the inlet feed tube and ambient temperature and pressure. Through the LabVIEW software, the results were collected.

The tested prototype was equipped with flanges that connected the mixer to the pipes. It was therefore possible to quickly adapt the prototype to mixer.



Fig. 4. Measurements of pressure drop in a laboratory test facility

For WALKER mixer, the experimental results of pressure drops were collected and used for calculating the coefficients (of viscous and inertial resistance for CFD simulations) based on the quadratic polynomial equation and compared with Darcy-Forchheimer equation, as it was presented in details in our previous study [8]. In the plot (Fig. 5) there are differences in pressure drops for tested mixer and a reference case without mixer.

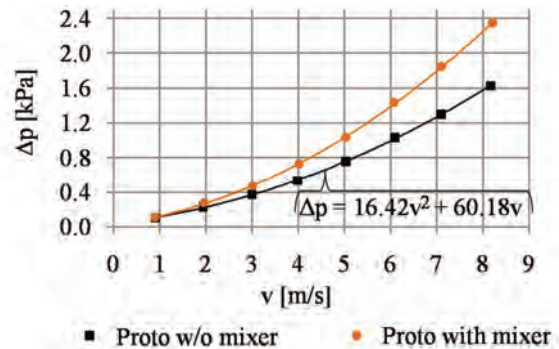


Fig. 5. Pressure drop vs. velocity for WALKER prototypes

Another test stand (Fig. 6), was used to study the velocity distribution at the monolith's outlet surface. The construction of the X-Y platform allows for automatic scanning of the entire cross-section area with a resolution of approximately 150 sampling points. The locally measured value by anemometer was recorded and the velocity profile was generated on this basis as it was in similar Tenneco Edenkoben study from Gehrlin et al. [5].

The laboratory test stand allows also to determine the NO_x reduction efficiency and urea conversion under steady state condition. In order to mimic as much as possible the given operating parameter of the engine in real time, the stand has electric heaters, NO_x gas dispenser, UWS dosing and sampling probe for CLD (chemo luminescence detector). In addition, it is equipped with mass flow, temperature and pressure sensors.

In this article, the conversion of NO_x will be omitted, only the distribution of ammonia on the outlet surface of SCR monolith will be presented in order to compare it with CFD simulation results. Further research, however, will take into account the experimental results of NO_x reduction efficiency and comparing them with numerical simulations.



Fig. 6. Measurements of flow distribution in a laboratory test facility

3. Reverse engineering

To determine the correct geometry of the 3D model for CFD simulation, reverse engineering process was used. Here, the reverse engineering should be understood as the process of obtaining a geometric CAD model from 3D points by scanning or digitizing existing part [8].

In the first phase of the process, the aftermarket prototype (AM) and mixer were scanned using a laser scanner (ROMER Absolute Arm). Noncontact scanning technology

allows to obtain a cloud of points from parts in Metrolog X4 software, which define the surface geometry.

The next step was loading the point cloud data (TXT format file) to the Leios 2 software. Cleaning of noise and reduction of the number of points were applied. Using built-in tools, triangle meshes and cross-sections as NURBS (Non-uniform rational B-spline) were generated.

Finally, based on the base data from Leios 2 (cross-sections, points, planes), 3D models in Catia v5 were re-constructed (Fig. 7).

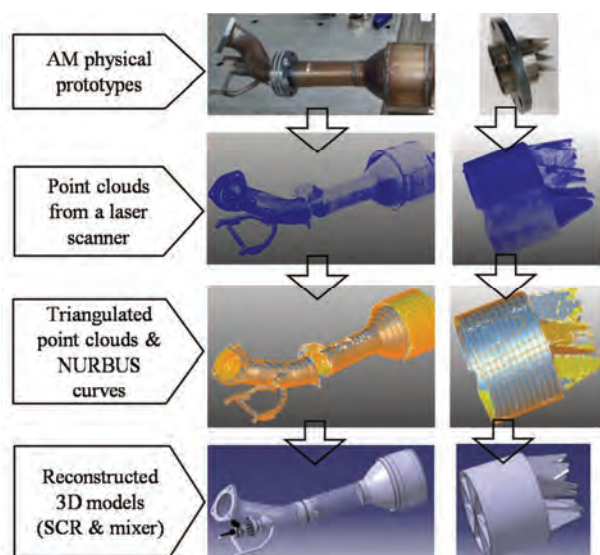


Fig. 7. Process of reverse engineering for AM prototype SCR and mixer

The reverse engineering for the original part (Fig. 8) was similar as in WALKER prototype. The exception was in VW mixer, in which the process ran differently due to not destroying parts (mixer cutout). Therefore, using the OLYMPUS IMPLEX NX videoscope, the original mixer was reviewed, and thanks to this, the 3D model of the mixer in Catia v5 program was reproduced as closely as possible.

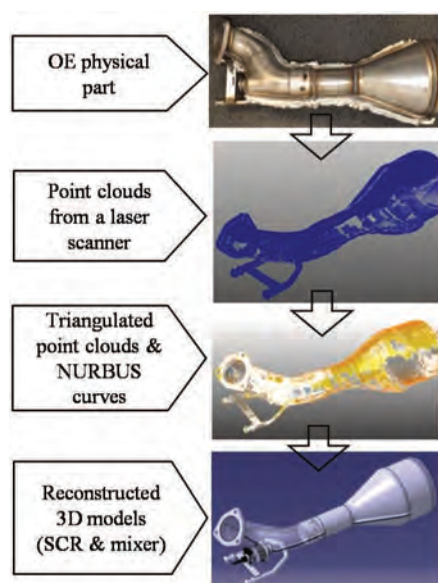


Fig. 8. Process of reverse engineering for OE SCR

4. CFD modeling

For CFD simulations, internal models of SCR with cut out mixer geometry had to be prepared. Models have been simplified in such a way that all the insignificant details (edges, fillets, grooves) were removed. This allowed for a better mesh adjustment (tetrahedral and hexahedral) with acceptable mesh quality (orthogonal quality, skewness etc.).

Steady state flow simulation through the SCR were performed. The governing equations for the flow within the domain are RANS (Reynolds Averaged Navier-Stokes) equations consisting of the conservation of mass, momentum and energy equation [8]. The flow inside the porous medium (monolith) was treated as laminar, whereas in the remaining volume of the flow domain the turbulence was modeled by means of the Realizable $k - \epsilon$ model.

Numerical simulation was carried out in two ways (Table 4). First, simulations were carried out only for ideal gas flow and ambient temperature ("cold flow"). This allowed for comparison of the CFD results with experimental tests (distribution of velocity fields and pressure drops).

In the second type of CFD simulation ("hot flow") also contains species transport with volumetric reactions by using laminar interactions between UWS and inflow of the mixture. In this case the DPM (Discrete Phase Model) was applied for three injections. Such settings allowed to check the degree of mixing and distribution of NH_3 in monolith.

Table 4. General settings for simulations in ANSYS Fluent

COLD FLOW – without UWS	
Analysis type	Steady state with 250 iterations
Turbulence model	κ -epsilon; realizable
Near-wall treatment	Standard wall functions
Working fluid	Air; ideal gas
Porous media	Laminar, porous with viscous & inertial coef.
Inlet conditions	Mass flow rate 115kg/h & ambient temp. 295 K
Outlet conditions	Open to atmosphere & ambient temp. 295 K
HOT FLOW – with UWS	
Analysis type	Steady state with 500 iterations
Turbulence model	κ -epsilon; RNG
Near-wall treatment	Standard wall functions
Species transport:	Mixture of urea-water-air
– Reactions	Laminar finite-rate interaction + diffusions
DPM	Continuous phase& Iteration + Latent Heat
Injection x3:	Solid-cone with multicomponent particle type
– Material	Urea-water (32.5% $\text{CO}(\text{NH}_2)_2$; 67.5% H_2O)
– Dia. distribution	Rosin-Rammler
– Temperature	293 K (UWS)
– Velocity	20 m/s
– Outer radius	60 μm (radius of injector hole)
– Cone angle	4° (included half-angle)
– Total flow rate	0.021 kg/h
– Min. diameter	4 μm (the smallest droplet diameter)
– Max. diameter	80 μm (the largest droplet diameter)
– Mean diameter	50 μm
– Spread parameter	3.5 (for a single-phase nozzle)
– No. of diameters	5 (no. of diameters in each distribution)
Vaporization model	Urea-water: convection/diffusion-controlled
Porous media	Laminar, porous with viscous & inertial coef.
Inlet conditions:	Mass flow rate 206 kg/h & temp. 677 K
– Species mass frac.	6.1% O_2 ; 7% H_2O ; 14.6% CO_2
Outlet conditions	Open to atm. (pressure outlet) & temp. 677 K
Walls: DPM	Boundary cond. type: reflect

The whole process of preparing a simulation model for the WALKER prototype and the original VW part, is briefly presented in Fig. 9 and Fig. 10.

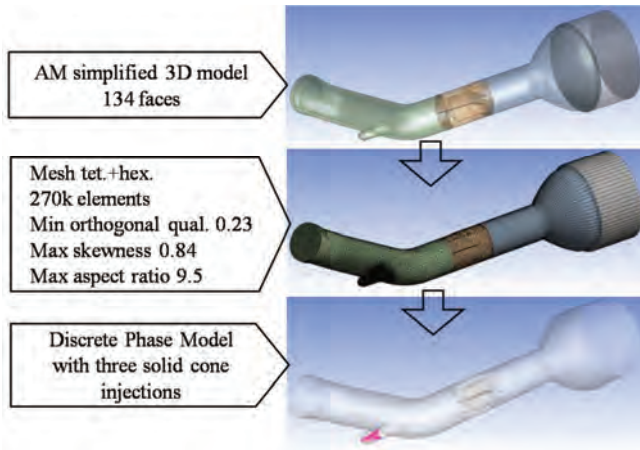


Fig. 9. Process of preparation of AM 3D model for CFD simulation

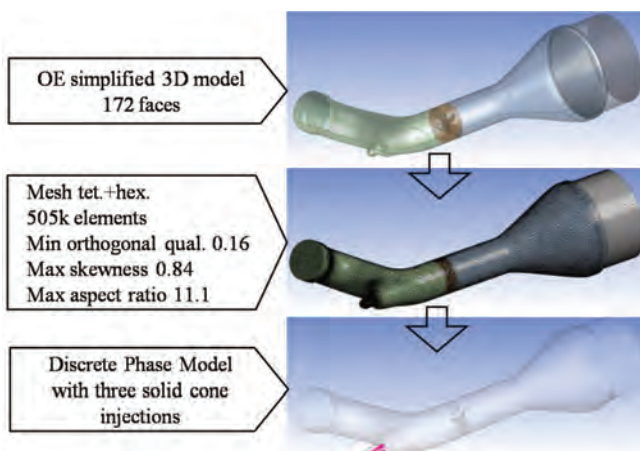


Fig. 10. Process of preparation of OE 3D model for CFD simulation

In ANSYS Fluent, the Rosin-Rammler distribution function is used to initialize droplets size distribution into the gas phase. The mass fraction Y_d of droplets of diameter greater than d can be given by [13]:

$$Y_d = e^{-(d/\bar{d})^n} \quad (2)$$

where: d – diameter, $\bar{d}$ – size constant (mean diameter), n – size distribution parameter (spread parameter).

The effect of the convective flow of the evaporating material from the droplet surface to the bulk gas phase (Stefan Flow) and the rate of evaporating mass (convection/diffusion controlled model) is given by [13]:

$$\frac{dm_p}{dt} = k_c A_p \rho \ln(1 + B_m) \quad (3)$$

where: m_p – droplet mass, k_c – mass transfer coefficient, A_p – droplet surface area, ρ – gas density, B_m – is the Spalding mass number given by:

$$B_m = \frac{Y_{i,s} - Y_{i,\infty}}{1 - Y_{i,s}} \quad (4)$$

where: $Y_{i,s}$ – vapor mass fraction at the surface, $Y_{i,\infty}$ – vapor mass fraction in the bulk gas.

The mass transfer coefficient k_c is calculated from the Sherwood number correlation [13]:

$$Sh = \frac{k_c d_p}{D_{i,m}} = 2.0 + 0.6 Re_d^{1/2} Sc^{1/3} \quad (5)$$

where: d_p – droplet diameter, $D_{i,m}$ – diffusion coefficient of vapor in the bulk, Re_d – Reynolds no., Sc – Schmidt no.

5. Results

Selected cases will be presented and compared with experimental and numerical results. Furthermore, the main flow parameters in the SCR system will be discussed.

To evaluate the velocity flow distribution, the uniformity index is commonly used. The uniformity index (UI) represents how a specified field variable varies over a surface, where a value of 1 indicates the highest uniformity. The area-weighted (captures the variation of the quantity) of UI is calculated using the following equation [13]:

$$\gamma = 1 - \frac{\sum_{i=1}^n [(\phi_i - \bar{\phi}) A_i]}{2|\bar{\phi}| \sum_{i=1}^n A_i} \quad (6)$$

where: ϕ – specified field variable, $\bar{\phi}$ – average value of the field variable, n – number of facets, i – facet index of a surface, A – surface area.

Uniformity index $\gamma \geq 0.94$ can be treated as a very good result and the value at this level is considered satisfactory.

Comparing the results (Fig. 11) of CFD and experiments, it can be concluded that the pictures of the velocity fields look similar. In the final results for that case (Table 5), UI for CFD is 0.961 while for experiment it is 0.952.

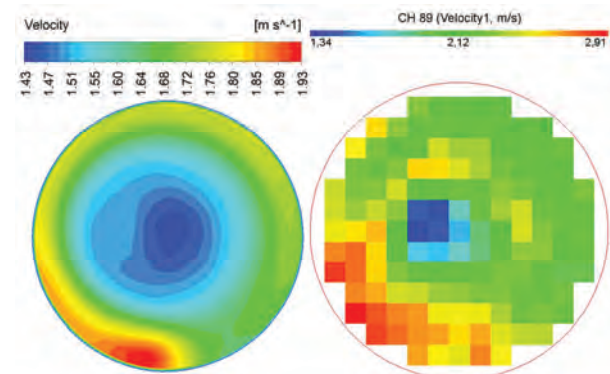


Fig. 11. Case 2: CFD vs. experimental (velocity distribution on the outlet)

Figure 12 shows sampling from the monolith outlet surface through zigzag movement from bottom to top.

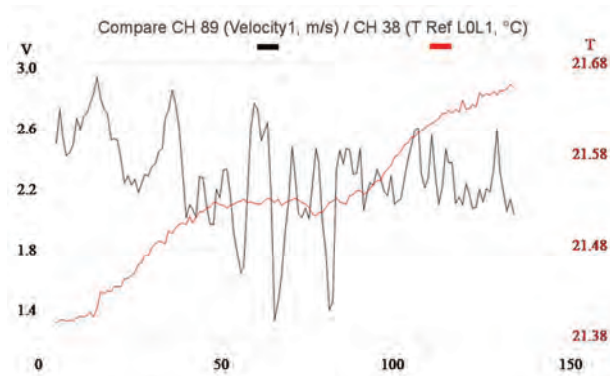


Fig. 12. Case 2: velocity and temperature distribution on the outlet

The next parameter for checking the flow in the SCR system is ETA index. ETA 10% should be > 0.7 whereas for ETA 20% value 1.0 is required. And for example, ETA 10% is area range with velocity $\pm 10\%$ of mean velocity, i.e. narrow range of areas ought to be avoided where there are large higher velocity compared to the mean velocity. However, ETA 15% (Fig. 13) has been checked for physically tested parts. For numerical simulations, ETA 15% limit has been implemented into ANSYS Fluent using User Defined Functions (UDFs) through Microsoft Visual Studio 2017 and C++ language.

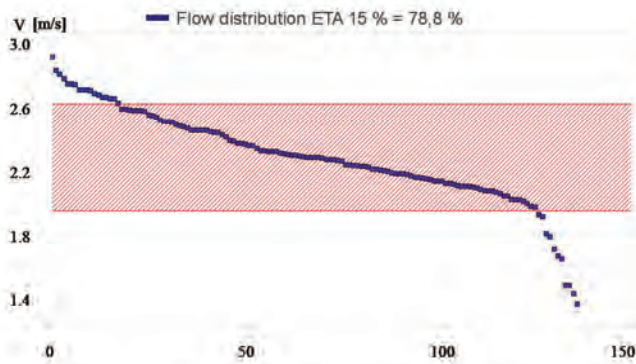


Fig. 13. Case 2: flow distribution (velocity) on the outlet (ETA 15%)

In turn, the Eccentricity index (7) determines the place on the monolith surface with the highest flow rate. It is desirable that such sites do not occur at the edge of the monolith, therefore this parameter should be ≤ 0.5 . Eccentricity is specified by the following equation:

$$\text{Eccen.} = \frac{r_{v_{\max}}}{R} \quad (7)$$

where: $r_{v_{\max}}$ – radius to the point of maximum velocity, R – radius to the edge of monolith.

The velocity streamlines in the SCR system (Fig. 14), illustrate the direction of gas mixing. In CFD, to determine pressure drops within the mixers, inlet and outlet facet average of static pressure (difference) calculated.

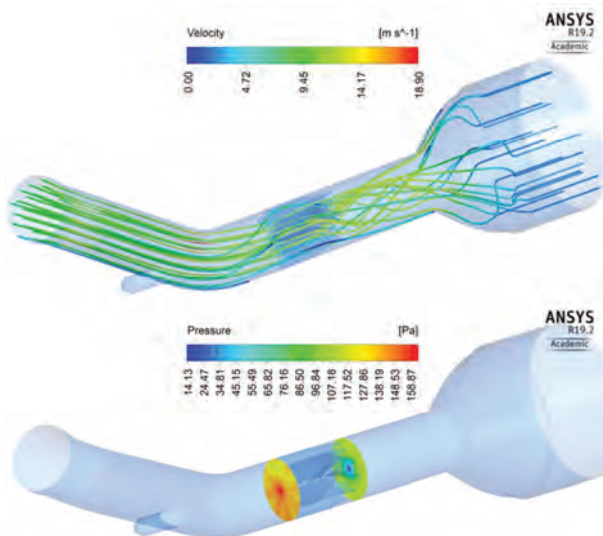


Fig. 14. Case 2: velocity flow and pressure drop through mixer

The pressures in the entire SCR system (Fig. 15) have been compared through volume rendering. The exhaust gas pressure drops over the whole exhaust system is equal to exhaust pressure at the manifold (backpressure). Excessive backpressure creates overmuch heat, lowers engine power, fuel consumption and damage of the engine parts. Therefore, backpressure in a certain level leads to improvement of the engine performance and reduces emissions [8].

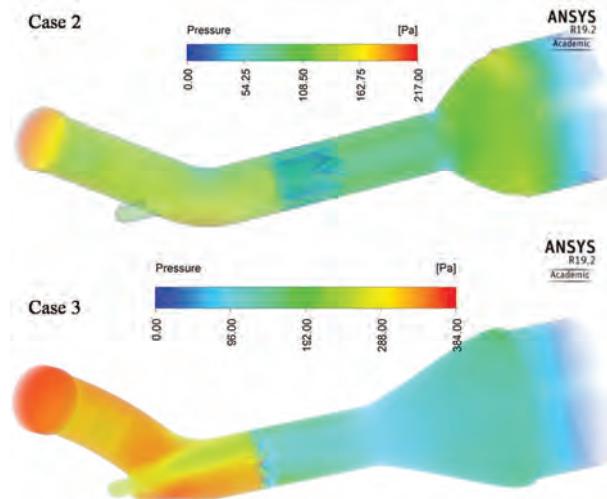


Fig. 15. Static pressure for the whole SCR

Subsequent simulations were carried out with an injection of UWS (Fig. 16). In this view, the case 2 has a greater use of the mixer at the bottom while for the VW part shows mostly distribution of UWS on the edges of the walls.

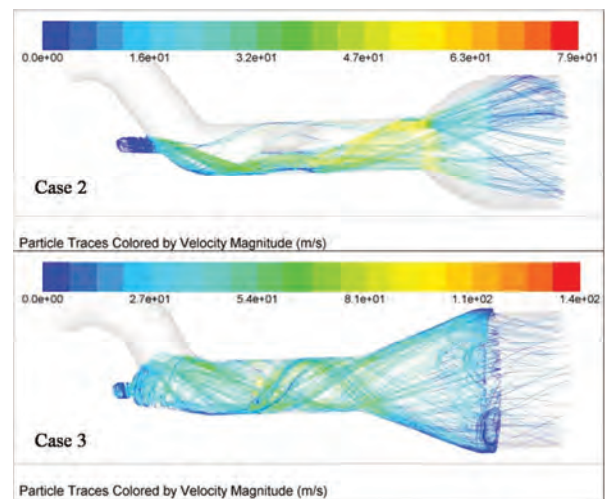


Fig. 16. Velocity particle trajectories of UWS through mixer

Mixture homogeneity is one of the most important parameters used in SCR systems. To evaluate the quality of mixing degree of ammonia with exhaust gases, the coefficient of variation (CoV) is used and defined as [14]:

$$\text{CoV} = \frac{\sigma}{\bar{x}} = \frac{1}{\bar{x}} \sqrt{\frac{\sum_{i=1}^n (x_i - \bar{x})^2}{n}} \quad (8)$$

where: σ – standard deviation, x_i – each of the values of the data, $\bar{x}$ – the mean of the x_i , n – the number of data points.

CoV is defined as the standard deviation of concentration (e.g. NH_3), over the mean concentration. If the value of CoV is equal to zero, it means that system is mixed completely and the efficiency of mixing operation is 100%. A large CoV means an uneven distribution of ammonia.

For previously presented cases (Fig. 16), the mixing of ammonia (mole fraction of NH_3) on the front surface of the monolith (Fig. 17) was also compared. At this point, mixing degree should be the best due to starting the chemical reactions with the SCR catalyst. In this case it turned out to be better WALKER mixer (NH_3 CoV = 0.60 and UI = 0.755).

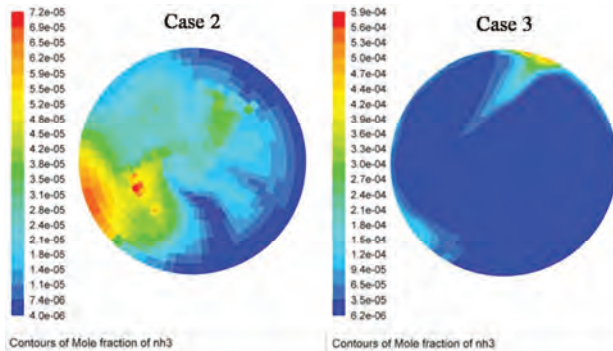


Fig. 17. Mole fraction of NH_3 on the inlet face of monolith

The undesirable phenomenon of “ammonia slip” on the monolith's outlet surface has also been compared (Fig. 18).

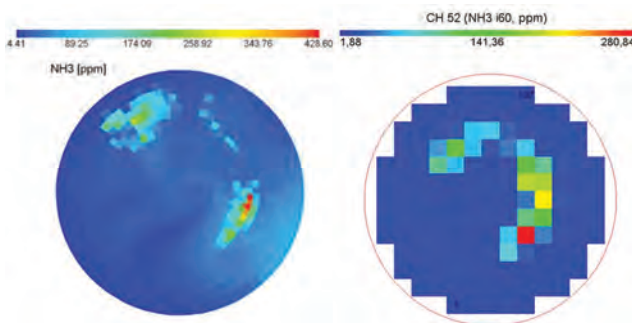


Fig. 18. Case 2: CFD vs. experimental (NH_3 on the outlet face)

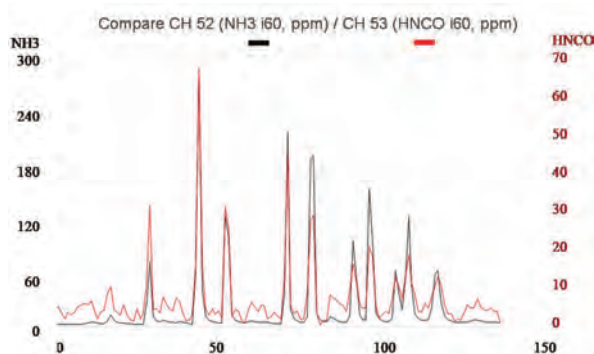


Fig. 19. Case 2: NH_3 vs. HNCO on the outlet face

The mole fraction of NH_3 were higher compared to the experimental results due to not taking into account the NO_x reactions. The presence of a large amount of NH_3 also causes presence of isocyanic acid (Fig. 19) which can lead to undesirable deposit.

The remaining results and their comparisons are presented in the final table (Table 5).

Table 5. Experimental and numerical results for the studied cases

Unit	w/o mixer	WALKER mixer	OE VW mixer
	Case 1	Case 2	Case 3
UI v. e.	N/A	0.952	N/A
UI v. n.	0.963	0.961	0.961
BP e., mbar	2.11	2.43	N/A
BP n., mbar	1.92	2.17	3.84
ΔP e., Pa	w/o	32	N/A
ΔP n., Pa	w/o	34	211
ETA 15% e.	N/A	78.8	N/A
ETA 15% n.	90.0	95.0	88.6
Eccen. e.	N/A	0.89	N/A
Eccen. n.	0.19	0.47	0.46
CoV NH_3	1.11	0.60	1.87
UI NH_3	0.552	0.750	0.647

e. – experimental results; n. – numerical results (CFD); v. – velocity;
 UI v. – uniformity index of velocity on the outlet surface of monolith;
 BP – max. static pressure in SCR; ΔP – pressure drop of the mixer;
 ETA15% – index of velocity on the outlet surface of monolith;
 Eccen. – Eccentricity of velocity on the outlet surface of monolith;
 CoV NH_3 – coef. of variation for NH_3 on the inlet surface of monolith;
 UI NH_3 – uniformity index for NH_3 on the inlet surface of monolith;
 N/A – result not available (experimental study not carried out).

6. Summary

The SCR system without mixer has smallest pressure drop, however, along with the complexity of the mixer construction, the pressure drop in the system increases but at the expense of this the uniformity index of NH_3 and degree of NH_3 mixing (CoV) are improved.

Although the WALKER and the original VW mixer had the same UI for velocity gas distribution, the WALKER mixer had better level of mixing (CoV) and UI of NH_3 distribution. Surprisingly, the mixer of the original part had poor level of mixing (NH_3 with highest concentration on the walls) and large pressure drop on the mixer.

High level of ammonia slip in the results, can be explained by the fact that the tests were performed only with the participation of the first SCR reactor, while the original part has two separate SCR reactors connected by a intermediate tube, thanks to which ammonia slip can be eliminated and the level of mixing even better. Simulations confirm this, because about half of UWS have evaporated, while it should be completely evaporated before reaching the monolith. This could also be achieving by reducing the gas flow rate, use a longer pipe or inlet cone for better mixing or using additional thermal insulation for SCR.

The position and rotation angle of the mixer in the SCR is important. When the droplets of UWS do not hit the mixer and just pass and hit the wall, there may be a risk of creating a deposit. Therefore, various attempts to set up the mixer should be used, so that the concentration of NH_3 was symmetrical after mixer and until reaching the monolith.

Small differences in the results between CFD simulations and experimental studies can be explained by the fact that the 3D models for numerical simulations have been simplified and thus the flow inside the SCR system is less disturbed than in the physical parts. Furthermore, the measurement uncertainties contribute to the differences as well.

The test results did not reach a very high level. Improvements regarding the construction of a static mixer and NO_x conversion studies will be the subject of further work.

Acknowledgements

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Nomenclature

CF	Conformity Factor	NURBS	Non-Uniform Rational B-Spline
CFD	Computational Fluid Dynamic	PEMS	Portable Emission Measurement System
CLD	Chemo Luminescence Detector	PM	Particulate Matter
CoV	Coefficient of Variation	PN	Particulate Number
DEF	Diesel Exhaust Fluid	RDE	Real Driving Emission
DPF	Diesel Particulate Filter	SCR	Selective Catalytic Reduction
DPM	Discrete Phase Model	SDPF	SCR-Catalyzed Diesel Particulate Filter
LNT	Lean NO _x Traps	UI	Uniformity Index
NEDC	New European Driving Cycle	UWS	Urea-Water Solution
NSC	NO _x Storage Catalyst	WLTP	World Harmonised Light Vehicle Test Procedure

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Assessment methods of the basic parameters of the combustion process in reciprocating internal combustion engines

The article presents selected methods of assessing the basic parameters of the combustion process, as well as assessing the usability and limitations of the methods used to determine the initiation and the end of the combustion process in reciprocating internal combustion engines. The methods considered are based on data contained in real, developed indicator diagrams. Basic thermodynamic assumptions and the scope of application of the combustion process evaluation method based on the actual work cycle of a combustion engine prepared in a double logarithmic scale were discussed. The article also mentions the application of the following methods: a direct pressure comparison method in the cylinder, the comparison of the first pressure derivative in the cylinder, logarithmic derivative method of pressure change in the cylinder, the method of the polytropic index, method of the first derivative of the polytropic index and the method of constant values of the polytropic index. The article presents the advantages and disadvantages of the research of our methods.

Key words: combustion process, combustion process parameters, indicator diagram, engine operation cycles

1. Introduction

The combustion process in a piston combustion engine is one of the basic processes of the engine's work cycle. Its assessment requires the use of appropriate research methods based on the analysis of the actual work cycle. In order to achieve the best possible effect of converting energy contained in fuel into mechanical work, it is necessary to precisely control work processes occurring in the engine cylinder [7]. An important issue in the above-mentioned area is the assessment and control of the combustion process. This topic is currently being addressed by many researchers around the world, and its complexity is related mainly to the number of independent factors directly affecting the efficiency and average theoretical pressure of the engine work cycle [1, 12]. There are many research methods to evaluate the engine cycles presented and widely described in the work of engineering researchers [7, 11, 15]. The use of new fuels, the pursuit of oil diversification and the European Union's policy of rational energy management in which transport is one of the key industries, requires increasing use of new generation fuels and assumes the widespread use of compressed natural gas CNG [16]. In Poland, supplying vehicles with CNG gas is still not common, although according to many specialists it is a promising fuel for the future [10]. This requires adapting currently operated and produced engines to supply these fuels and contributes to the need to become more familiar with the process creation of combustible mixtures and the process of their combustion. This entails the need to develop and analyse combustion process evaluation methods that can be used in the process of further engine development, mainly in terms of steering, playing a key role in achieving the required energy effect and maintaining the lowest possible concentration of harmful exhaust components. [13]. The evaluation methods of the combustion process provide information on many indicators that play an important role in the evaluation of the engine control, which can also be used to diagnose intra-cylinder processes [8, 9]. The effect of thermodynamic, thermochemical and heat exchange processes occurring in the piston cylinder of the internal combustion

engine are noted on an indicator diagram. It is a quantitative and qualitative source of information about these processes. Its analysis makes it possible to determine engine indices, heat dissipation characteristics during the combustion process, the composition of the working medium as a function of the angle of rotation of the crankshaft and many others. Its course also affects the noise of the engine. The course of the curve of the indicator graph depends primarily on the quality and course of the atomisation process of the fuel having a decisive impact on the quality of the fuel-air mixture and the aerodynamic properties of the air and the way it is fed into the cylinder [4]. The developed indicator diagram of a reciprocating internal combustion engine is the basis for the application of the described methods for assessing the combustion process parameters.

Modern design solutions for motor control and regulation systems enable modelling of component work cycle processes in a wide range [3, 6]. Differences in physical and chemical properties of modern fuels, including biofuels and gaseous fuels, dictate the need for a thorough assessment of the impact of feeding them on the characteristics of their operation. A slight deviation from the stoichiometric composition of the combustible mixture contributes significantly to the emission of harmful exhaust components [13]. The possibility of precise dosing of gaseous fuel and the maintenance of good atomisation determines the correct course of combustion of the fuel-air mixture [14]. The paper presents and characterises several methods of combustion process evaluation based on the developed indicator diagram. The basic parameters of this process are the beginning and end of the combustion process and the duration of the combustion process.

2. Preparation of a real indicator diagram for analysis

Evaluation of the combustion process parameters based on the actual indicator diagram requires obtaining reliable values of the pressure course in the engine cylinder. The uniqueness of the work cycles obtained consecutively during the registration of the pressure course as a function of the angle of rotation of the crankshaft requires the initial

processing of data consisting of averaging the experimentally recorded diagrams. According to [2] the minimum number of work cycles should not be less than 33. Current data archiving technology and computer software enable the recording of a pressure signal with a frequency of 0.1° of crankshaft rotation ($^\circ\text{CR}$) and a record of a much higher number of subsequent pressure courses. Preparation for the analysis of the experimentally removed indicator graph must include an accurate and reliable determination on the indicator diagram of the position of the piston in the upper returnable position (URP) [2]. The chart obtained in this way is subjected to further processing enabling determination of the beginning and end of the combustion process on its basis and the duration of the combustion process depending on the analysis method used. The introduction of additional results determining the moment of fuel injection in the case of compression ignition engines, or the determination of the spark moment on the spark plugs in the case of forced ignition engines enables to obtain information on the duration of the self-ignition delay period and the duration of the ignition delay period. The results obtained may be subject to an error caused by high-frequency noise in the form of a hum. The use of signal filtration or smoothing methods distorts the information originally received.

3. Characteristics of methods for analysing combustion process parameters

There are many methods to assess the parameters of the combustion process in a reciprocating internal combustion engine. Basic methods of evaluation of combustion process parameters include the measurement method with the use of optical methods to determine the moment of the appearance and location of the flame, and methods based on the analysis of the actual indicator graph recorded during the experiment. In the next points of the article from 3.1 to 3.7 a description of seven selected methods for the assessment of the basic parameters of the combustion process based on the analysis of the indicator graph is presented:

- a) method of analysing the combustion process on the basis of a graph prepared in a double logarithmic scale
- b) a direct pressure comparison method in the cylinder
- c) the method of comparing the first pressure derivative in the cylinder dP
- d) logarithmic derivative method of pressure change in the cylinder $d\ln P$
- e) the method of the polytropic index k
- f) method of the first derivative of the polytropic index dk
- g) the method of constant values of the polytropic index k_p

The method of evaluation of combustion process parameters based on direct comparison of pressure in the engine cylinder (point 3.2) and direct comparison of the first pressure derivative in the engine cylinder (point 3.3) does not allow determining the end of the combustion process, thus it is not possible to use these methods to determine the duration of the combustion process. In the method of analysing the combustion process, two cases were considered on the basis of an indicator chart prepared in a double logarithmic scale. The first describes the combustion process parameters in the event of the start of the combustion process before the upper reverse position of the piston,

while the second describes a special case during operation of the engine with the initiation of the combustion process after the upper reverse position of the piston. A graphical interpretation of the described cases is shown in Figure 1a and 1b, described in 3.1. Some methods require complex calculations, and interpretation of the obtained results is difficult. The problems are particularly visible during the analysis using dk and k_c methods that are very sensitive to noise. All of the above-mentioned methods are based on the analysis of the results obtained when the pressure in the engine cylinder is indicated. The calculations are carried out using a previously prepared, averaged indicator diagram.

3.1. Method of analysing the combustion process on the basis of a graph prepared in a double logarithmic scale

The evaluation of the combustion process parameters using the indicator diagram prepared in a double logarithmic scale requires the assumption that the engine cylinder is a closed thermodynamic system, i.e. there is no bleeding of the working medium through the leakage of the piston – rings – cylinder assembly and no leakage at the inlet and outlet valves. The assumption makes it possible to treat intra-cylinder processes in the form of polytropic compression transformations and expansion described by the equation [5]:

$$pV^n = B = \text{const.} \quad (1)$$

where: p – pressure, V – volume, n – polytropic index.

To the above equation, dimensionless values of pressure $\bar{p}$ and volume $\bar{V}$ were introduced by actions in order to be able to mathematically interpret the obtained values [1]:

$$\bar{p} = \frac{p}{p_0} \quad \text{and} \quad \bar{V} = \frac{V}{V_c} \quad (2)$$

where: p_0 – ambient pressure, V_c – volume of the combustion chamber, p , V – instantaneous values of cylinder pressure and volume.

Using the obtained dimensionless values in equation (1), and then logarithmising and ordering them, the equation of a straight line was obtained:

$$\log \bar{p} = -n \log \bar{V} + C \quad (3)$$

The form of equation (3) indicates that the value of the polytropic transformation index is the tangent of the angle of the straight line to the axis of the abscissa of the logarithmic system. The processes of compression and expansion of the real indicator chart presented in the double logarithmic scale allow to determine the value of indexes of polytropic transformations. The beginning of the combustion process will be determined by the point of detachment of the curve of logarithms of dimensionless pressure values in the cylinder from the determined straight line for the case of initiation of the combustion process before TDC (Fig. 1), or the point of intersection of the logarithm curve with this straight line in case of ignition after the upper return position of the TDC piston (Fig. 2).

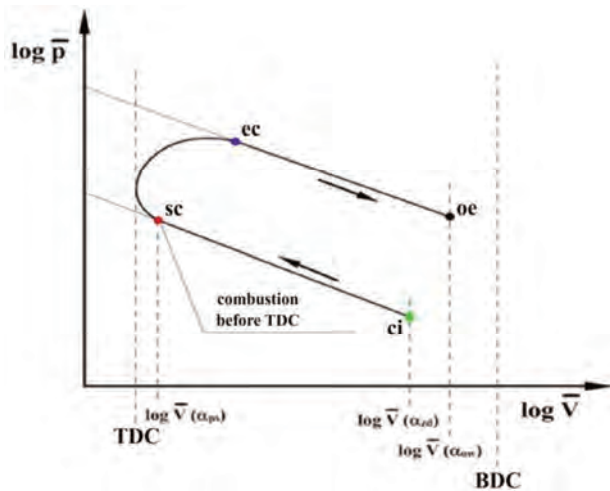


Fig. 1. Graphical representation of the combustion process evaluation method takes into account the occurrence of ignition before top dead centre: sc – the beginning of the combustion process, ec – the end of the combustion process, ci – closing the intake valve, oe – opening the exhaust valve

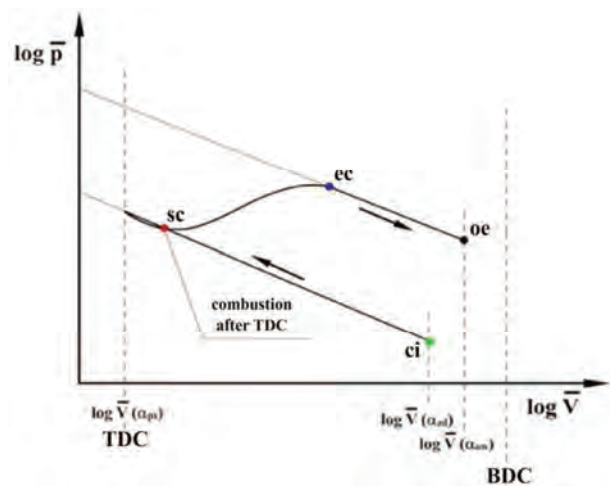


Fig. 2. Graphical representation of the combustion process evaluation method takes into account the occurrence of ignition after top dead centre: sc – the beginning of the combustion process, ec – the end of the combustion process, ci – closing the intake valve, oe – opening the exhaust valve

3.2. Direct pressure comparison method in the cylinder

The direct pressure comparison method in the cylinder is one of the test methods using measurement data obtained during the engine's indication. It is used to determine the moment of initiation of the combustion process and ignition delay. However, it is not possible to determine the duration of the combustion process with this method. Determination of the initiation moment of the combustion process is defined as the point of rapid deviation of the pressure curve of the cycle with combustion from the curve of the cycle pressure without combustion. An illustration of the method is shown in Fig. 3.

This method requires the creation of pressure courses in the cylinder both for the combustion engine cycle and without combustion. Specialised equipment of the station is required to enable a cycle without combustion. The results obtained in this way for a cycle without combustion may not be meaningful to those obtained for the combustion cycle. The desirability of obtaining results for a cycle with-

out combustion using the required equipment is an interference in the operation of the engine. The comparative analysis of the values of pressure waveforms from the combustion and non-combustion cycle may be biased because the results obtained from two significantly different motor states are compared. Noise can cause a scattering of pressure waveforms that make comparisons difficult. Therefore, the introduction of the criterion of differences in the values of compared pressures was adopted. The comparative criterion for the P method is in the range of 3% to 10%. According to the 3% criterion, the occurrence of ignition is considered when the value of the pressure difference of the cycle with combustion and without combustion reaches 3%.

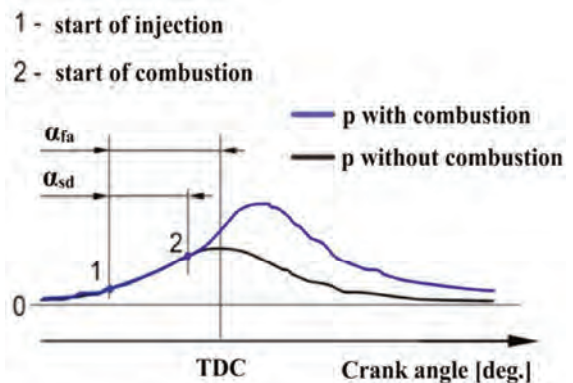


Fig. 3. Graphical representation of the direct pressure comparison method in the cylinder: α_{fa} – fuel injection advance angle, α_{sd} – self-ignition delay angle

3.3. Method of comparing the first pressure derivative in the cylinder dP

The self-ignition delay period in this method is determined based on a comparison of the first pressure derivative during the combustion cycle and the first cycle pressure derivative without combustion. The beginning of the combustion process is determined by the deviation of the first derivative of the pressure change of the cycle with combustion from the curve representing the values of the first derivative of the cycle pressure without combustion. Optionally, the second pressure derivative in the cylinder of the engine can be used for calculations. A graphic illustration of the method is shown in Fig. 4.

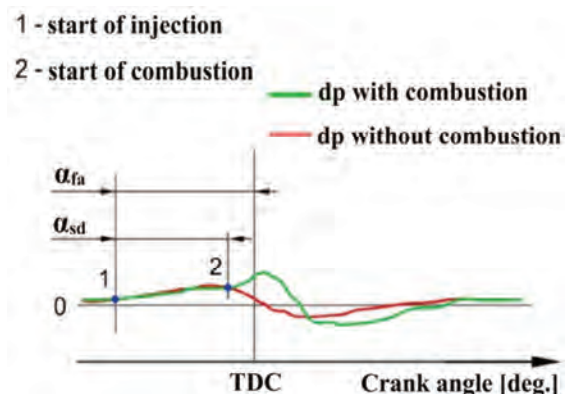


Fig. 4. Graphical representation of the method of determining the self-ignition delay based on the course of the first pressure derivative in the cylinder: α_{sd} – fuel injection advance angle, α_{fa} – self-ignition delay angle

As in the P method, also in this it is required to have a pressure course in the cycle cylinder without the combustion process. The disadvantage of the method is the inability to determine the end of the combustion process.

3.4. Logarithmic derivative method of pressure change in the cylinder $d\ln P$

The idea of the method is based on the use of a logarithmic derivative of the course of pressure change in the cylinder of the engine. The beginning and the end of the combustion process is determined by means of two characteristic points of the logarithmic derivative of a pressure change, which are two local minimal values of the $d\ln P$ curve prepared as a function of the angle of rotation of the crankshaft. The course of the value of this curve is described by the equation:

$$\frac{d(\ln p)}{d\alpha} = \frac{\ln p_{i+1} - \ln p_i}{\alpha_{i+1} - \alpha_i} \quad (4)$$

where: p , α – dimensionless quantities.

In Figure 5 a graphic illustration of the method is presented.

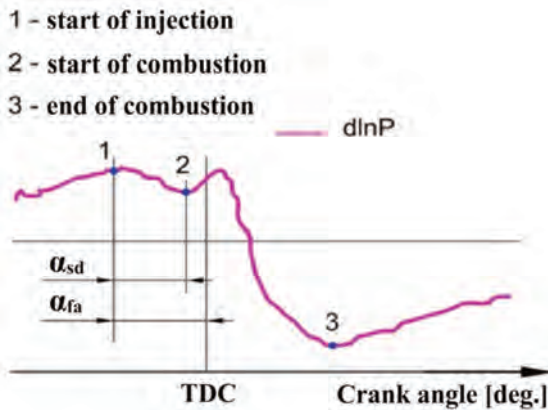


Fig. 5. Graphical interpretation of determination of combustion process parameters based on the logarithmic derivative of the pressure change in the cylinder of the engine: α_{pw} – fuel injection advance angle, α_{axis} – self-ignition delay angle

In this method, to determine combustion parameters, it is not required to have a pressure course in the cylinder from the cycle without combustion. The $d\ln P$ method allows to determine both the beginning and the end of the combustion process. The disadvantage of the method is the large spread of data. For large ignition angles, this method gives results with a large error, because the first peak of the minimum that marks the beginning of combustion disappears. The intensity of noise when measuring for large ignition angles also makes it difficult to find a characteristic local minimum, indicating the initiation point of the combustion process, also introducing errors in determining the ignition delay and duration of the combustion process.

3.5. Method of the polytropic index k

In this method, the determination of combustion parameters is based on the value of the polytropic index. The values of the polytropic index are determined by the formula:

$$k = - \frac{\frac{dp}{p}}{\frac{dV}{V}} \quad (5)$$

where: p – pressure in the engine cylinder, V – volume of the working medium in the cylinder, dp – the first pressure derivative in the cylinder, dV – the first derivative of the cylinder volume.

In order to determine the combustion parameters, the polytropic indexes are compared for the combustion engine cycle and without combustion. The beginning of the combustion process is determined by the point of propagation of the curves, while the end of the combustion process determines the point of re-joining the curves. This method is shown in Fig. 6.

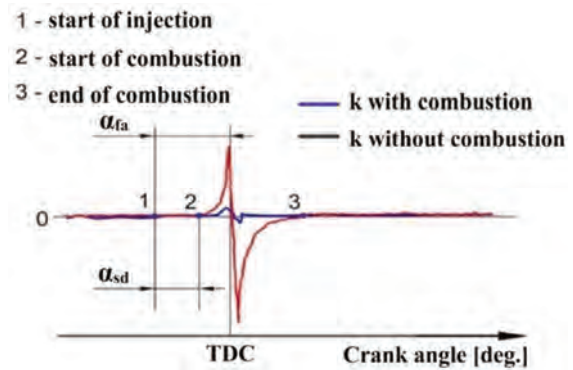


Fig. 6. Graphical interpretation of determination of combustion process parameters using the polytropic index k : α_{fa} – fuel injection advance angle, α_{sd} – self-ignition delay angle

The disadvantage of this method is the large influence of the quality of the measurement data on the final result. This is mainly because the differential function is used in the calculation algorithm. Differentiation is a mathematical action that exposes differences in input values. Therefore, the seemingly gentle course after the differential operation reveals, and to a significant degree, all kinds of minimal disturbances, e.g. noise in the input signal.

3.6. Method of the first derivative of the polytropic index dk

The method is based on data obtained as a result of the engine's indication. They are used to determine the course of the value of exponent polytropic compression k . This method uses the derivative of the compression polytropic index. The formula used to determine the value of the polytropic index has the following form:

$$k_i = \frac{\log \frac{p_{i-1}}{p_i}}{\log \frac{V_{i-1}}{V_i}} \quad (6)$$

where: p_{i-1} – the actual value of the pressure in the engine cylinder for the angle of rotation of the crankshaft α_{i-1} , p_i – the actual value of the pressure in the engine cylinder for the angle of rotation of the crankshaft α_i , V_{i-1} – the calculated cylinder volume for the crank angle rotation α_{i-1} , V_i – the calculated cylinder volume for the crank angle rotation α_i .

The values of the polytropic index, obtained on the basis of formula (6), are then subjected to differentiation operations:

$$dk(\alpha) = \frac{dk_i}{d\alpha} \quad (7)$$

The result of equation (7) is the course of the compression polytrope as a function of the angle of rotation of the engine crankshaft. On the basis of the course obtained, it is possible to determine the period of self-ignition delay. The method is used to determine the boundaries of the combustion process in the engine. As the points of beginning and end of the combustion process, we assume moments for which the $dk/d\alpha$ curve takes the value of zero. This method does not require to have a polytropic index on the compression cycle for the cycle without combustion, i.e. no pressure cycles of the compression process are necessary. A graphic illustration depicting the described method is shown in Fig. 7.

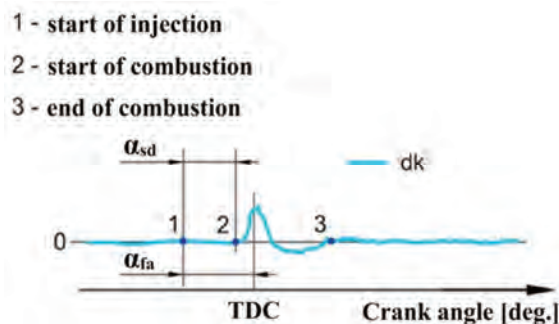


Fig. 7. Illustration of the method for evaluation of combustion process parameters using the first derivative of the polytropic index: α_{fa} – fuel injection advance angle, α_{sd} – self-ignition delay angle.

3.7. Method of constant values of the polytropic index k_p

The determinant of the beginning and end of the combustion process are the breakaway points of the curve, respectively $\frac{dk}{d\alpha}$ from a constant value (initiation of the combustion process) and its re-adoption of a constant value (end of the combustion process). As in the previously discussed methods, also in this we use the results obtained in the process of indication of the engine (the actual value of the pressure in the cylinder and the corresponding angle of rotation of the crankshaft). In the Fig. 8 the method described graphically is presented.

For different values of the ignition advance angle, the collected data is characterised by quite a large spread. This creates difficulties in determining the detachment points and the return to the horizontal straight line, which takes a constant value, which are next to the beginning and the end of the combustion process.

Conclusions

The evaluation of the combustion process in a piston combustion engine is a very complex problem and is still

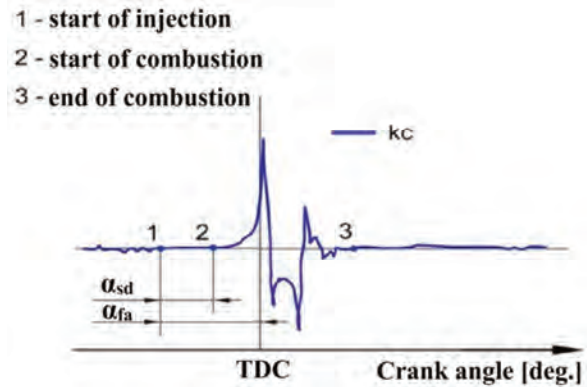


Fig. 8. Graphical representation of the method for determining the beginning and ending of the combustion process based on fixed values of the polytropic index, allowing the determination of the self-ignition delay: α_{fa} – fuel injection advance angle, α_{sd} – self-ignition delay angle

a field of interest for engineers' researchers and is the subject of numerous scientific and research works. Constant efforts to reduce the harmful impact of engines on the environment dictated by the restrictions of exhaust emission standards and rationalisation of intra-cylinder processes aimed at achieving the greatest possible conversion of chemical energy contained in fuel requires adjusting tools to assess the basic parameters of processes in the engine cylinder. These are the methods of evaluation of one of the most important components of the work cycle, i.e. the combustion process, which primarily influences the formation of harmful exhaust components in the cylinder and plays a key role in the efficiency of the engine. Among the methods described, we can distinguish those that require engine cycles with combustion and without combustion, we include the method of direct comparison of cylinder pressure and the method of comparing the first pressure derivative in the cylinder. These simple methods allow to determine only the moment of initiation of the combustion process and determination of the ignition delay period of the combustible mixture. The methods of the polytropic index, the constant value of the polytropic index, and the first derivative of the polytropic index, are sensitive to the level of noise. The limitation in the application of the logarithmic derivative method of pressure change in the cylinder of the $d\ln P$ engine is engine operation at high ignition angles and combustion of poor mixtures. The methods of assessment of the basic combustion process parameters presented in the article show advantages and disadvantages, therefore their use must be accompanied by the researcher's intuition and reliable evaluation and verification of the results obtained by means of several available research methods.

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An experimental analysis of performance and exhaust emissions of a CRDI diesel engine operating on mixtures containing mineral and renewable components

The manuscript presents a comparative analysis of the performance and emission characteristics of a compression ignition engine equipped with a Common Rail injection system. The engine is fueled with diesel-biodiesel mixtures containing 25% and 50% share (by volume) of renewable components. Conventional diesel is used as a reference. Turkey lard and rapeseed oil are used as raw materials and subjected to the single-stage transesterification process to obtain methyl esters. The experiments are performed on a medium-duty, turbocharged, inter-cooled, Common Rail Direct Injection (CRDI) diesel engine. This study concentrates on one engine speed of 1500 rpm, typical for gen-set applications, and mid-load range from 100 Nm to 200 Nm. The scope of measurements covers the analysis of exhaust gasses concentration and engine efficiency parameters. In addition, the in-cylinder pressure measurements are performed in order to provide insight into the differences in combustion characteristics between examined fuel mixtures. The study reveals that the addition of the renewable component to fuel mixture positively affects a number of examined performance parameters as well as decreases the concentration of the examined toxic exhaust components, in the majority of cases.

Key words: exhaust emissions, common rail, biodiesel, animal fat biodiesel, combustion analysis

1. Introduction

The current stage of technological development still requires combustion engines as the source of propulsion for numerous vehicles and machines. However, due to the vast greenhouse gas (GHG) footprint in combination with diminution of reserves, it is obvious that dependence on energy derived from fossil fuels should be limited. As a result of extensive research, many alternative energy carriers have been developed as a substitution of conventional diesel fuel (DF) [1]. Currently, one of the most promising paths for biofuel production involves the transesterification of raw fatty material with alcohol [2]. Biodiesel, as a product of transesterification, is non-toxic, biodegradable and directly applicable in compression ignition engines (CI) as a stand-alone fuel, as well as an admixture to DF. Biodiesel can be obtained from a wide range of raw materials including plant oils and animal fats (edible, non-edible or waste quality).

The use of methyl esters causes a number of more or less desirable consequences for engine performance and emissions. Regardless of the origin (plant or animal), the amount (stand-alone fuel or admixture to DF) or type of the engine used (single- or multi-cylinder, equipped with modern or former type injection system) biodiesel is likely to cause reduction of exhaust emission indexes like hydrocarbons and carbon monoxide. On the other hand, the main disadvantages related to biodiesel combustion are the increase in specific fuel consumption and extended emission of oxides of nitrogen [3–5].

In terms of combustion characteristics, biodiesel brings similar effects when compared to conventional DF. However, some significant differences were reported in common rail (CR) engines adopting sequential injection schemes [3, 6].

The aim of the present study is to identify the differences in performance and emission parameters, caused by the use of different fuel composition, in a modern, multi-cylinder, CI engine operating on a standard (factory) con-

troller. The discussion is supported by the results of in-cylinder pressure measurements.

2. Materials and methods

The study concerns biodiesel mixtures produced in the laboratory conditions. Raw fatty materials used for biodiesel production are turkey lard and rapeseed oil. Biofuels are produced by a single-step transesterification reaction of the raw material with methyl alcohol, performed in the presence of an alkaline catalyst (potassium hydroxide), at ambient pressure. The laboratory equipment, as well as reaction parameters, are identical to those described in previous studies [3, 6] (alcohol to fatty material molar ratio is 6:1, the reaction temperature is $60^{\circ}\text{C} \pm 2^{\circ}\text{C}$, the reaction is carried for 45 minutes). Other activities related to biofuel production concern removal of residual alcohol, glycerol phase, solid impurities, and moisture.

Obtained fatty acid methyl esters (FAMES) are admixed to DF in volumetric proportions such that the fuel mixture contains 25% or 50% of renewable components. Mixtures of the turkey lard methyl esters are labeled as T25 and T50, respectively. Corresponding mixtures of the rapeseed oil methyl esters are labeled as R25 and R50.

Physicochemical properties of the fuel mixtures and DF are determined according to the procedures specified in PN-EN 14214 Standard. The analysis includes the examination of selected parameters: density at 15°C , viscosity at 40°C , sulfur content, water content, total contamination and cold filter plugging point (CFPP). The results of the analyzed physicochemical properties are listed in Table 1.

Performance and emission tests are performed on a 2.6 dm^3 , four-cylinder, Common Rail CI engine with Bosch EDC16C39 control unit. The relevant test engine data is presented in Table 2. Remaining elements of the test stand are based on AVL devices (dynamometer: Dyno Perform 240; engine speed control system: THA100; fuel balance: 735S; air mass flow meter: SENSYFLOW P; emission

bench: AMA i60; test stand control and data acquisition system: PUMA Open). The in-cylinder pressure measurement system is provided by KISTLER and coupled with an in-house acquisition and post-processing software, based on the LabVIEW environment. The association of pressure signal with precise crankshaft position is provided by an optical encoder. The in-cylinder pressure signal is recorded with the resolution of 0.1 crank angle degrees (CA). A more comprehensive description of the test stand and procedures for performed experiments is presented in [7, 8].

Table 1. Physicochemical properties of the analyzed fuels

Sample	DF	T25	T50	R25	R50
Density at 15°C [kg/m ³]	828	836	849	839	852
Viscosity at 40°C [mm ² /s]	2.80	3.08	3.37	3.15	3.52
Sulphur content [mg/kg]	6.1	4.41	3.37	4.74	4.09
Water content [mg/kg]	61	150	244	126	172
Total contamination [mg/kg]	9	13.6	17.4	11.6	15.8
CFPP [°C]	-12	-12	-8	-12	-12

Table 2. Technical data of the engine

Manufacturer	Andoria Mot
Engine type	ADCR
Number of cylinders / arrangement	4/in line
Fuel injection system	Common Rail, Bosch CR2.0
Displacement volume	2636 cm ³
Rated power / rotational speed	85 kW/3700 rpm
Max. torque / rotational speed	250 Nm/1800–2200 rpm

For each test run, the engine rotational speed and load are stabilized with the absolute accuracy of ± 5 rpm and ± 2 Nm, respectively. After the stabilization, steady-state measurements are performed, under predefined engine rotational speed $n = 1500$ rpm and the following load conditions: 100 Nm, 150 Nm and 200 Nm. The same measurements are performed for each of the tested samples.

Basic operational parameters of the engine: generated power, fuel, and air consumption and temperatures, as well as in-cylinder pressure and injector coil current, are recorded during the experiments. To expose the differences of efficiency between individual fuel mixtures – brake fuel conversion efficiency (BFCE) was also calculated as a relation between mechanical work performed by the engine and the amount of energy introduced with the fuel.

The concentrations of the hydrocarbons and carbon monoxide are recorded, with the relative accuracy of less than 1% of the maximum range of the respective analyzer.

3. Results and discussion

In the following subsections, the results of BFCE and exhaust gasses concentration, for specific fuel mixtures, are presented as relative values with respect to the diesel reference. The in-cylinder pressure and the injection coil current diagrams in subsection 3.2, present data averaged over 100 consecutive cycles.

3.1. Fuel conversion efficiency results

Figure 1 presents the differences in the BFCE for fuel mixtures containing 25% biocomponents, under the examined operating conditions. Figure 2 show analogous for mixtures with 50% share of biocomponents.

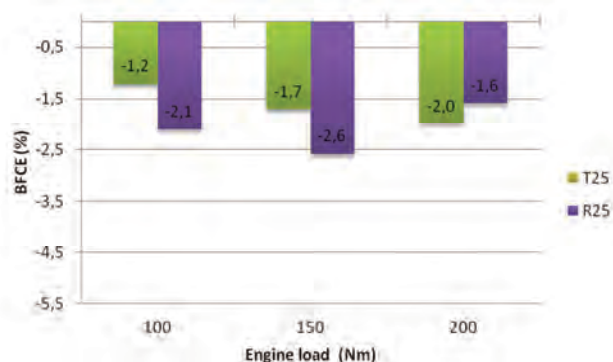


Fig. 1. Changes of BFCE in relation to DF for T25 and R25 under different engine load conditions

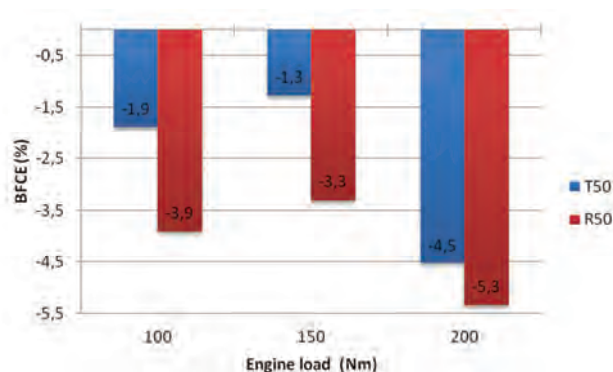


Fig. 2. Changes of BFCE in relation to DF for T50 and R50 under different engine load conditions

Presented results indicate that the biodiesel obtained from turkey lard is characterized by better conversion efficiency than biodiesel obtained from rapeseed oil. The exception is the 200 Nm case (Fig. 1), where the R25 mixture presents better fuel efficiency than T25. Note, however, that the difference between both biofuels does not exceed 0.4 percentage point which is below the level of significance at this specific operating point.

The relation between biodiesel content and fuel efficiency is additionally noted. Mixtures containing higher biodiesel content have been characterized by bigger differences in fuel efficiency in relation to DF, especially under high load conditions (Fig. 2).

3.2. In-cylinder pressure results

Figure 3 to Figure 5 present recorded in-cylinder pressure and injection coil current for different loads and fuel mixtures.

The injection actuation signal indicates that for all operating conditions and fuel mixtures, the injected fuel value is divided into two doses – the pilot and the main dose. For both biofuel mixtures, the pilot injection for engine operating conditions of 150 Nm and 200 Nm is significantly advanced from the DF reference (Fig. 4 and Fig. 5). This phenomenon is observed regardless of the biodiesel content in the fuel mixture. Under the engine load of 150 Nm and 200 Nm, the injection of the main fuel dose for examined biodiesels is also performed at an earlier stage of the cycle than for DF. This effect is particularly manifesting for mixtures containing biofuel obtained from turkey lard.

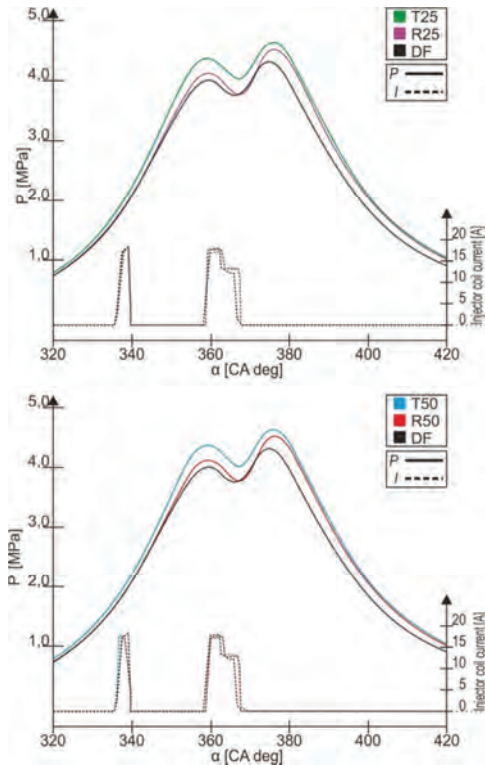


Fig. 3. In-cylinder pressure and injection current versus CA for 100 Nm load

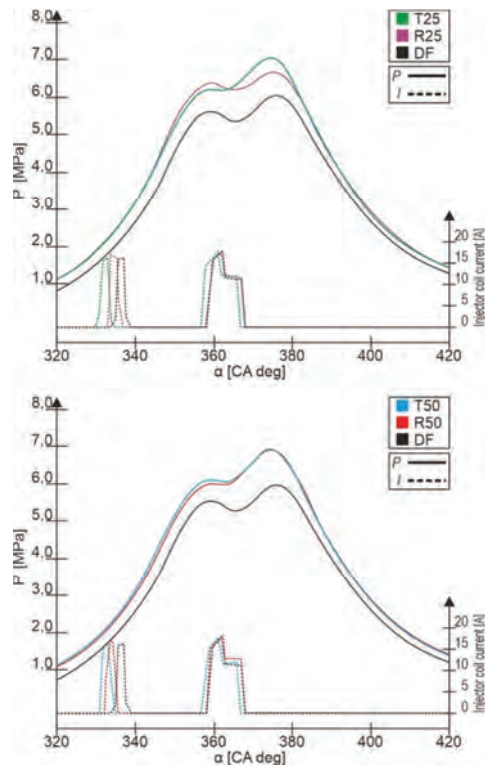


Fig. 4. In-cylinder pressure and injection current versus CA for 150 Nm load

The differences in injection timing for different fuel compositions are introduced by the EDC as a response to differences in fuel properties (physicochemical presented in Table 1, as well as those influencing combustion process – resulting from differences in the molecular structure of the

mineral and renewable fuel components). Since the EDC controller is optimized for the conventional DF operation, it is difficult to indisputably evaluate the correctness of its operation. However, the differences in injection timing for individual fuel mixtures are reflected in in-cylinder pressure plots.

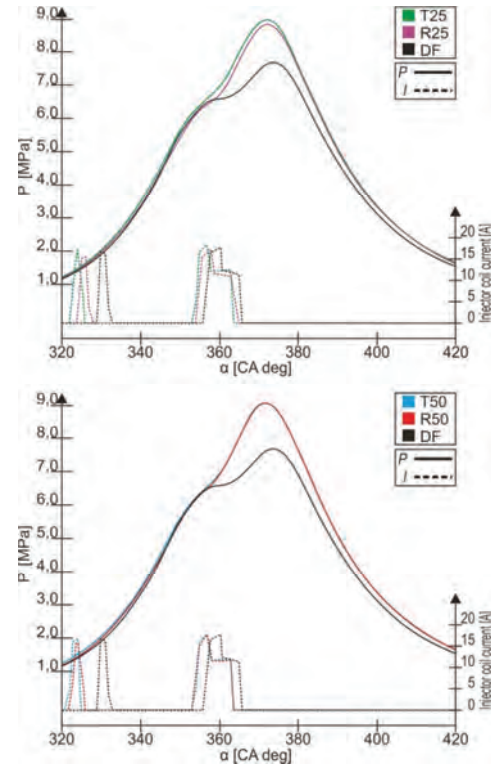


Fig. 5. In-cylinder pressure and injection current versus CA for 200 Nm load

For experiments conducted under 100 Nm load conditions (Fig. 3), mixtures T25 and T50 generate the highest pressure values in the entire analyzed range of the operating cycle. Consequently, the maximum pressure values for fuel mixtures containing both types of admixed biocomponent share significantly higher compared to those recorded during DF operation. This phenomenon is more visible during the experiments performed under the highest examined load (Fig. 5), where T50 and R50 present almost identical values during the whole analyzed part of the cycle (significantly higher from DF). The in-cylinder pressure traces for other fuel mixtures are the reflection of injection timing and highlight the similarities in the combustion behavior of respective biodiesels, despite different feedstock used for their production.

The 150 Nm case (Fig. 4) shows significant differences in the pressure signal between DF and biodiesels. The differences in the in-cylinder pressure occur during the whole analyzed part of the operating cycle. Globally lower pressure values recorded for DF, compared to biodiesels are a result of differences in the amount of air introduced into the combustion chamber. Therefore an increased degree of supercharging can be associated with the use of renewable fuel components at some engine operating points. This is most probably controlled by the EDC (by means of variable

geometry turbocharger) since increased fuel dosing is applied due to the de-rated heating value of the biofuels. The EDC attempts to maintain the lambda at a value designated by the engine map, thus applying higher boost pressures corresponding to elevated fueling.

As a general remark, the higher indicated pressures in the expansion part of the cycle, observed for both biofuels, suggest that higher indicated work is generated in order to sustain the same brake engine load. This indicates consistently higher friction for FAMES, most probably caused by increased parasitic losses in fuel injection equipment. Those are related to increased fuel viscosity and lower heating value (Table 1). The increased friction would explain the lowest BFCE for rapeseed oil-based biodiesel sample which ultimately score worst, in both fuel parameters (viscosity and LHV), compared to the T25 and T50 samples (see Fig. 1 and Fig. 2). The above thesis is to be checked in a dedicated experimental endeavor,

3.3. Exhaust gasses concentration results

Due to the significant discrepancies in the amount of the air introduced into the combustion chamber, during DF and biodiesel operation (Fig. 4), the 150 Nm case has not been discussed in the terms of the exhaust gasses concentration. The postulates based on such differences in engine operating conditions between the individual fuels could lead to misleading conclusions.

Figure 6 and Figure 7 present hydrocarbons concentration, in relation to DF for, specific fuel mixtures. Figure 8 and Figure 9 present the corresponding carbon monoxide emission results.

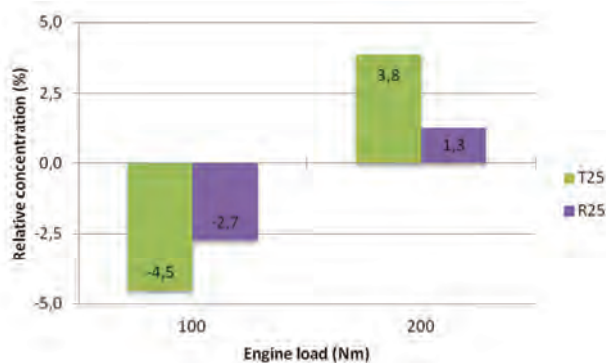


Fig. 6. Hydrocarbons concentration in relation to DF for T25 and R25

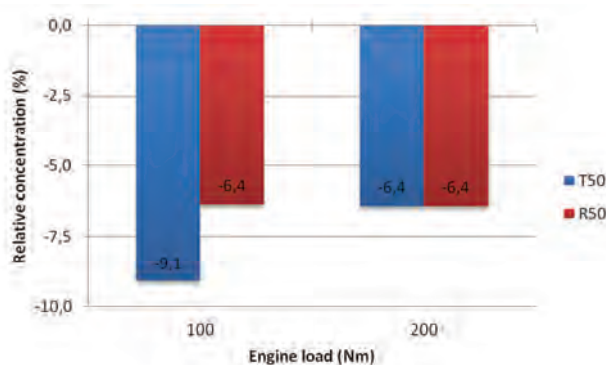


Fig. 7. Hydrocarbons concentration in relation to DF for T50 and R50

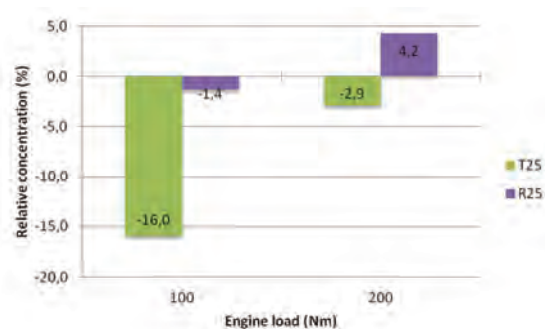


Fig. 8. Carbon monoxide concentration in relation to DF for T25 and R25

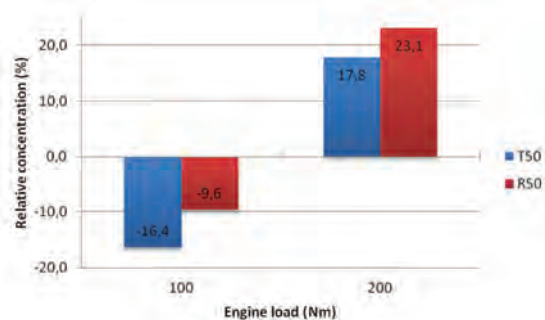


Fig. 9. Carbon monoxide concentration in relation to DF for T50 and R50

For the experiments conducted under 100 Nm, the concentrations of hydrocarbons and carbon monoxide are lower compared to DF, regardless of the origin or the quantity of the admixed biocomponent. Significantly better results are obtained for fuel mixtures containing animal origin biocomponents. Since biofuel mixtures containing animal-based FAMES presented significantly better efficiency results (Fig. 1 and Fig. 2) than plant origin ones, such results are expected. Furthermore, bigger differences are obtained for fuel mixtures containing higher amounts of admixed biocomponents.

During the experiments carried out under 200 Nm load, an increase of carbon monoxide concentration is noted (Fig. 8 and Fig. 9), with the exception for the T25 mixture, for which this emission index decreases slightly. However, in this operating point, the T25 notes a significantly higher concentration of unburned hydrocarbons than the R25 sample (Fig. 6). The reduction of hydrocarbons, along with the increase of carbon monoxide for T50 and R50 (Fig. 7 and Fig. 9) can be explained by significantly earlier injection (thus longer combustion duration) of biodiesel mixtures, compared to DF reference (Fig. 5).

4. Conclusions

In the present paper, the engine applicability of turkey lard methyl esters and rapeseed oil methyl esters mixtures are evaluated and compared with diesel fuel. During the course of the study, it is shown that a higher amount of admixed biocomponent reduces fuel efficiency. The biofuels obtained from animal origin raw material generally result in lower deterioration than plant origin components.

The respond of the electronic diesel control unit of the tested engine to biocomponent admixture is manifested in the acceleration of injection timing corresponding to the

biocomponent share. The use of biodiesel mixtures results in elevated maximum in-cylinder pressure values. In addition, an increased degree of supercharging can be associated with the use of renewable fuel components at some engine operating points.

In terms of exhaust gasses concentration, it is concluded that the increasing share of biocomponents in the fuel mixture causes greater differences in relation to diesel fuel. The use of biodiesel mixtures with 50% of admixed biocomponents results in a significant reduction of hydrocarbons and

carbon monoxide concentrations, especially at the lowest examined engine load conditions. Therefore the 200 Nm load cases generate a lower concentration of hydrocarbons together with increased concentration of carbon monoxide in relation to diesel fuel, consistently for both examined renewable components.

Of all the analyzed fuels containing renewable components, mixtures with animal origin FAMES, present the most favorable results in terms of efficiency and exhausts concentration.

Nomenclature

BFCE	brake fuel conversion efficiency
CA	crank angle
CFPP	cold filter plugging point
CI	compression ignition
CRDI	common rail direct injection
DF	diesel fuel
EDC	electronic diesel control unit
FAME	fatty acid methyl esters
TDC	top dead center

R25	fuel mixture containing 25% of rapeseed oil methyl ester and 75% of diesel fuel (by volume)
R50	fuel mixture containing 50% of rapeseed oil methyl ester and 50% of diesel fuel (by volume)
T25	fuel mixture containing 25% of turkey lard methyl ester and 75% of diesel fuel (by volume)
R50	fuel mixture containing 50% of turkey lard methyl ester and 50% of diesel fuel (by volume)

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Measurement of flame temperature and soot amount for effective NO_x and PM reduction in a heavy duty diesel engine

To reduce exhaust NO_x and smoke, it is important to measure flame temperature and soot amount in combustion chamber. In diesel combustion it is effective to use the two-color method for the measurement of the flame temperature and KL factor, which is related with soot concentration. The diesel flame was directly and continuously observed from the combustion chamber at running engine condition by using a bore scope and a high-speed video camera.

The experimental single cylinder engine has 2.0-liter displacement and has the ability with up to five times of the boost pressure than the naturally aspirated engine by external super-charger. The devices of High Boost, Wide Range and High EGR rate at keeping a relatively high excess air ratio were installed in this research engine in order to reduce exhaust NO_x emission without smoke deterioration from diesel engines.

The video camera nac GX-1 was used in this study. From observed data under the changing EGR rates, the flame temperature and KL factor were obtained by the software of two-color method analysis. The diesel combustion processes are understood well by analyzing high-speed movies of the diesel flame motion and its temperature. The NO_x and smoke are mutually related to maximum flame temperature and also it is possible to reduce simultaneously both NO_x and soot emissions by high EGR rate in a single cylinder diesel engine.

Key words: Diesel engine, exhaust emissions, NO_x, PM, EGR, BSFC, flame temperature, two-color method, video camera

1. Introduction

Today, as future power train of passenger cars EV (Electric Vehicle), FCV (Fuel Cell V), PHV (Plug in Hybrid V) and HV (Hybrid Vehicle) are researched and developed. However, the diesel engines are commonly used as power sources for ships, railcars, trucks, buses, passenger cars and other vehicles because they are superior to other engines in terms of fuel economy, durability, and reliability. They have shown improved exhaust emissions and thermal efficiency through high pressure injection [1–4], high boost with a turbocharging and intercooler [5–8], and enhanced EGR [9, 11]. Their high thermal efficiency can suppress CO₂ emissions and many passenger cars employ diesel engines in Europe. In spite of their many advantages, the reduction of exhaust emissions and exhaust PM are strongly desired [11–13]. Especially, these reductions are needed not only in regulation but also in real world [13].

The diesel combustion consists of fuel injection into the high pressure and high temperature of compressed air and thereby it is forming and burning an air-fuel mixture instantly in combustion chamber. For this reason, the diesel combustion yields a high flame temperature in the vicinity of spray and at the core of spray a flame temperature is low because of a lack of air. Much NO_x is yielded in the vicinity of spray and in high flame temperatures. On the contrary the smoke is yielded at spray core and here is relatively low flame temperature and fuel rich. To resolve these phenomena, the flame temperature measurements have been made using flame observations by high speed camera and an analysis by a two-color method [14–25].

In this paper author measures flame temperatures and KL factor, which is closely related with soot amount, on the cases of changing EGR rates and changing fuel injection pressures and studies not only the relation between the flame temperature and BSNO_x but also the relation between soot concentration and KL factor. K means a soot

concentration and L means an optical length. Now it is difficult to separate K and L.

2. History of temperature measurement

2.1. Shift to digital image data from color film

The author studied a flame temperature and KL factor by the two-color method since 1970s [14] and showed a good relation not only between a flame temperature and an amount of NO but also between the amount of soot and KL factor by gas sampling [15]. Figure 1 shows the one of the gas sampling result and flame temperature T (dashed line).

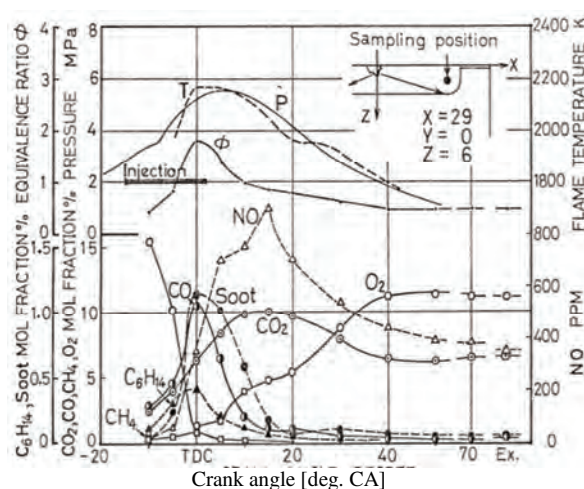


Fig. 1. Gas sampling results in comparison with flame temperature

Ahn [16] demonstrated the flame temperature measurement procedure using the two-color method with color film images of the diesel flame. The conventional mechanical high-speed film cameras have been replaced by electronic high-speed digital cameras (hereinafter designated as high-speed digital cameras), which can digitize and record image

data. The diesel flame temperature can be estimated from digital data obtained with a high-speed digital camera using the two-color method with a personal computer (PC) [17].

The temperature measurement from digital flame image by using the two-color method with a PC enables us to display easily a flame temperature distribution in a combustion chamber that had conventionally required complicated procedures to estimate by a film, and a KL factor distribution related to a smoke amount. This method contributes to NO_x and smoke reduction in exhaust emissions of a diesel engine.

2.2. Combustion observation

Conventional observation methods of diesel flame have used a combustion chamber with a large quartz observation window. These methods include the lower observation method by which a piston bottom is observed from the lower part of a combustion chamber with tempered quartz glass were conducted [18, 19]. Figure 2 shows the example of combustion photo. The upper observation method by which observation is conducted from the upper part of a cylinder head [20], and the side observation method by which or head [21, 22].

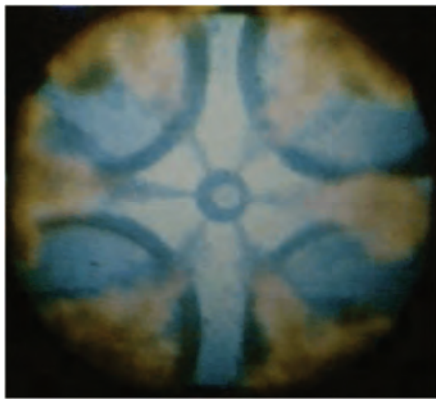


Fig. 2. Combustion photo by the lower window observation

Novel techniques included with the borescope method (by AVL proposal) were conducted by Shiozaki et al., which is combined with flame temperature measurement by the two-color method [23].

3. Analysis for flame temperature

3.1. Borescope

The author has adopted the improved borescope method, in which a 10 mm diameter is used for an optical probe (borescope) for observation of flame movement in the combustion chamber, and a diesel flame in the combustion chamber is watched through the observation hole in cylinder head, as presented in Fig. 3 [24, 25]. The observation range (red circle) in the piston combustion chamber is a part of sprays and is shown in Fig. 4. This method requires little engine modification, uses the same valve layout as the actual condition, and employs a real piston. Consequently, a diesel flame that is almost equivalent to that of a real system is visible. Because the borescope head protrudes from the underside of the head, it requires some machining to avoid interference with the piston cavity edge.

The machined portion is too small to affect combustion and the compression ratio ($\epsilon = 16.0$).

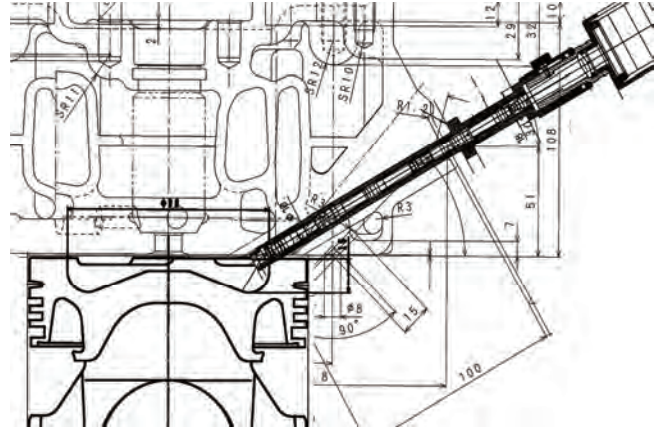


Fig. 3. Cross section of piston combustion chamber and the improved bore scope

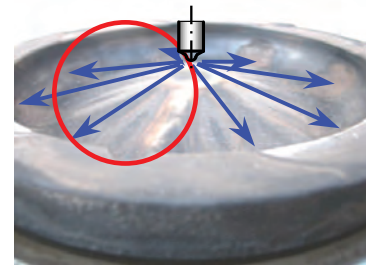


Fig. 4. Observation area in combustion chamber

In this study a high-speed digital camera (GX-1; NAC Image Technology Inc.) was used, which enables us to record images up to 200,000 fps. This study was conducted with 14,400 fps.

3.2. Two color method

Flame temperature measurements were conducted based on the diesel flame temperature analysis method using the two-color method, particularly addressing the brightness of soot particles reported by Kamimoto [14], using the flame temperature distribution analysis software developed by Tate [17] (hereinafter designated as flame temperature software).

A standard light unit for the two-color methods was used for acquiring the diesel flame temperature using flame temperature software. The calibration data for luminance and brightness temperature were shown in Fig. 5 and were

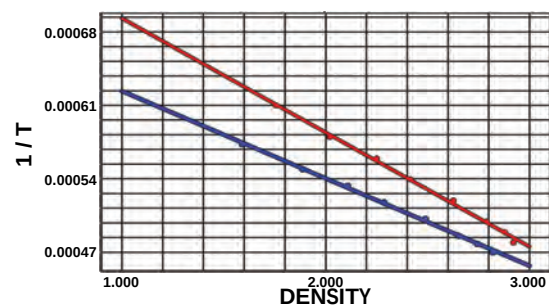


Fig. 5. Calibration data between the density and 1/T

measured by using this standard light unit. The absolute temperature scale was adopted for these calibration data in this report. The distribution of the diesel flame temperature and KL factor was obtained from the diesel flame images using the flame temperature software.

The representative value of flame temperature T_b is the average of temperatures at pixels where a flame is observed in the whole range of view. As reference Fig. 6 shows a spectral response of digital camera [17]. In this study the colors of red (610 nm) and blue (480 nm) were chosen.

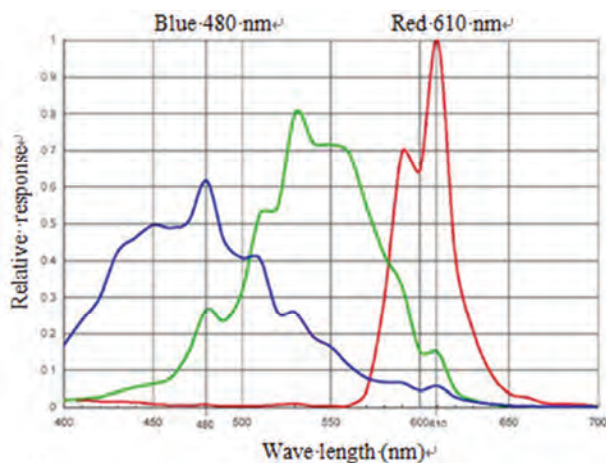


Fig. 6. Spectral response of digital camera between wave length (nm) and relative response

4. Experiment

4.1. Single cylinder test engine and system

This experiment employs a direct fuel injection diesel engine in a single-cylinder and four stroke cycle with boost pressure up to 600 kPa is applicable with an external motor-driven supercharger [9]. Figure 7 presents a system diagram of the research single-cylinder engine. It has a common rail injection system with fuel injection pressure up to 220 MPa, a mini sac nozzle, a nozzle hole diameter of 0.177 mm, eight nozzle holes, and a nozzle hole angle of 150°. The swirl ratio R_s generated from a cylinder head port was set constant as 1.4. A steel monotherm piston was used, for which the cross sectional view is presented in Fig. 3. Table 1 presents specifications of the single cylinder research engine.

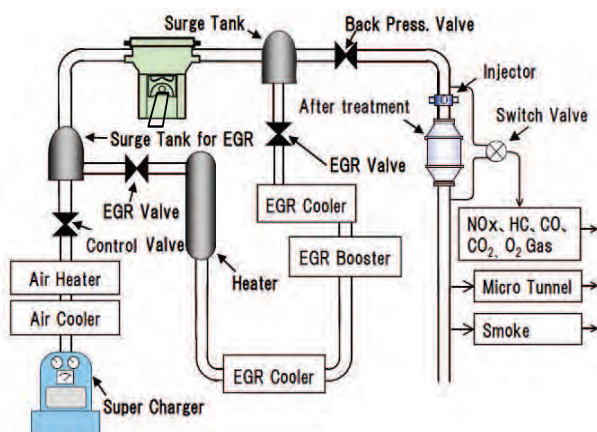


Fig. 7. Schema of experimental apparatus

Table 1. Engine specifications

Item	Specifications
Engine type	DI single cyl., 4 Valve
Displacement	2.004 L
Bore × stroke	135×140 mm
Max engine speed	2000 rpm
Injector	Common Rail system (Max P_{inj} = 220MPa)
Nozzle	Minisac ϕ 0.177×8-150°
Piston type	Steel piston (monotherm)
Comb. chamber	Toroidal (ϕ 98 mm)
Compression ratio	16.0
Swirl ratio	1.4
Valve actuation	Cam-less VVA
Air charging system	External super charger

Table 2. Fuel properties for the test

Category	Properties	Category	Properties
Density 15°C	g/cm ³ 08268	Elements	C 86.1
Kinematic	mm ² /s	mass %	H 13.8
Viscosity 30°C	3978		O —
Flash Point	°C 69.0		N <0.1
Cetane Index (JIS K2280)	61.0	Components	Saturates 83.4
Cetane Number	57.1	vol %	Olefins 00
			Aromatics 16.6
Distillation	IBP 175.5		Mono- 15.4
	5% 205.5		Di- 1.0
	10% 222.5		+Tri- 02
	50% 286.5	Calorific Value	kJ/kg 45900
	90% 332.5	Lower Calorific Value	kJ/kg 43030
	EP 354.5		
Sulfur	mass ppm 6	HFR (WS1.4)	μ m 233

4.2. Experimental conditions and variables

Experimental conditions are described as follows. Boost pressure P_b = 271.3 kPa and head swirl ratio R_s = 1.4 were fixed as common, and fuel injection pressure P_{inj} = 200 MPa and EGR rate = 50% were set as the standard conditions at engine speed N_e = 1,200 rpm. The brake mean effective pressure BMEP = 1.0 MPa (indicating a mean effective pressure IMEP = 1.2 MPa around), and fuel amount a stroke was q = 135 mm³/st. The following factors were varied.

The EGR rate was determined from the measurement of intake CO₂ concentration when exhaust CO₂ was mixed with the intake.

- The EGR rate was varied as 0, 30, 50, and 55% with the fuel injection pressure (Common-rail pressure) P_{inj} was fixed constant as 200 MPa.
- Fuel injection pressure P_{inj} was varied as 90, 150, 200, 220 MPa with the EGR rate fixed constant at 50%.

In the experiments the oxygen concentration of intake air containing EGR gas was measured at inside of the intake manifold, exhaust pressure was adjusted to the same pressure as the intake pressure P_b , and engine intake air temperature T_{in} was set to 50 ± 1°C. These conditions were the same as performance and exhaust emission tests.

5. Experimental results and discussions

5.1. Exhaust emissions and BSFC by changing EGR rate and injection pressure

Figure 8, 9, and 10 respectively present changes in exhaust emissions, fuel consumption performance (BSFC), and the excess air ratio λ when the EGR rate was varied as 0, 30, 50, and 55% and the fuel injection pressure (common rail pressure) P_{inj} was varied as 90, 150, 200 and 220 MPa.

Figure 8 shows that BSNO_x is reduced drastically from 21 g/kWh to 0.5 g/kWh at $P_{inj} = 220$ MPa, whereas BSHC and BSCO change little at a low factor, when EGR rate is increased from 0% to 55%. Smoke tends to increase by increment in the EGR rate. It increases only at $P_{inj} = 90$ MPa in the case of EGR rate = 50%, but it shows no increase at $P_{inj} = 150$ MPa and higher. Smoke then jumps up at EGR rate = 55%, but there is a little increase in smoke at $P_{inj} = 200$ and 220 MPa. It can be effective to reduce an amount of soot and PM by high pressure injection such as 200 MPa level.

Figure 9 suggests that BSFC varies a little by the EGR rate increase, and BSFC is minimum at $P_{inj} = 220$ MPa in

the case of EGR rate = 50%. Accordingly, when the EGR rate is enhanced to 50%, the fuel consumption rate reduces a little as fuel injection pressure is raised. Although P_{max} varies a little with the increase in the EGR rate, it rises as the fuel injection pressure increases. The pressure diagram changes a little because the air capacity of the cylinder is about the same even if the EGR rate is increased. However P_{max} rises as the fuel injection pressure increases because the peak of rate of heat release (ROHR) shifts to earlier timing. The NO_x concentration is expressed in parts per million (ppm) and reduces gradually with increasing EGR rate.

Figure 10 shows the intake CO_2 concentration, excess air ratio λ , intake O_2 concentration, and other results at an enhanced EGR rate. The excess air ratio λ is reduced from 4 to 2 as the EGR rate increases from 0% to 50%.

5.2. Increasing EGR rate

Figure 11 shows ROHR at EGR rates = 0, 30, 50, and 55% for injection pressure $P_{inj} = 200$ MPa. The photos of combustion at each crank angle after the ignition starting period of TDC are displayed in Fig. 12.

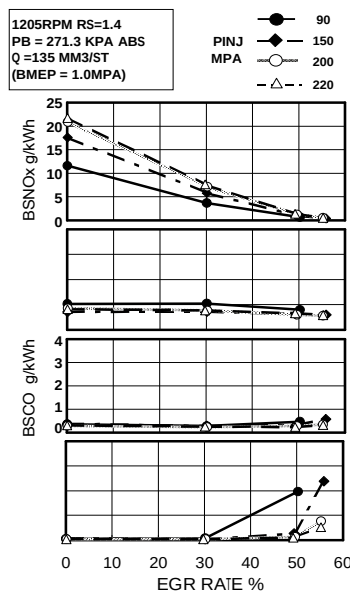


Fig. 8. Emission by changing EGR rate

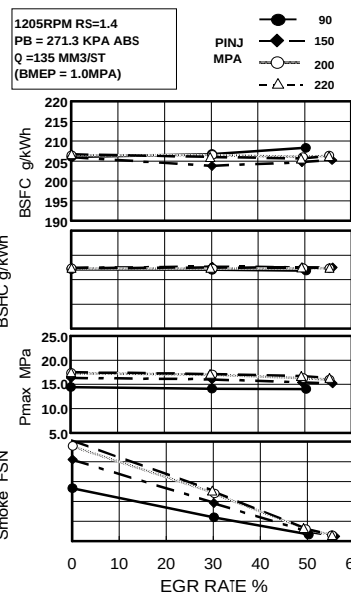


Fig. 9. Efficiencies by changing EGR rate

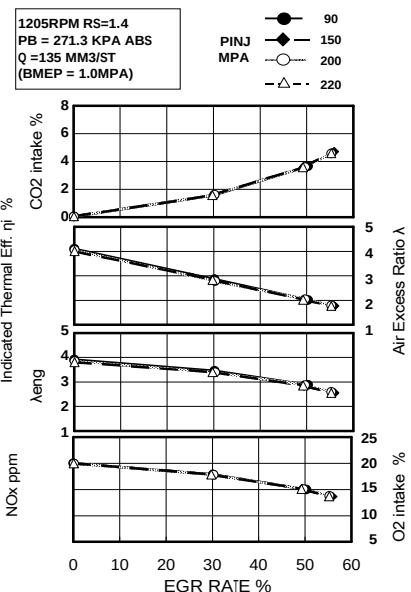


Fig. 10. Excess air ratio by changing EGR rate

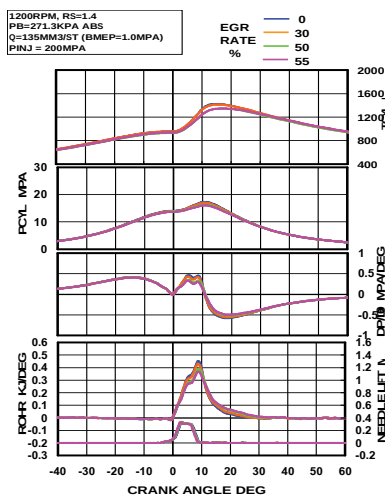


Fig. 11. ROHR by changing EGR rate

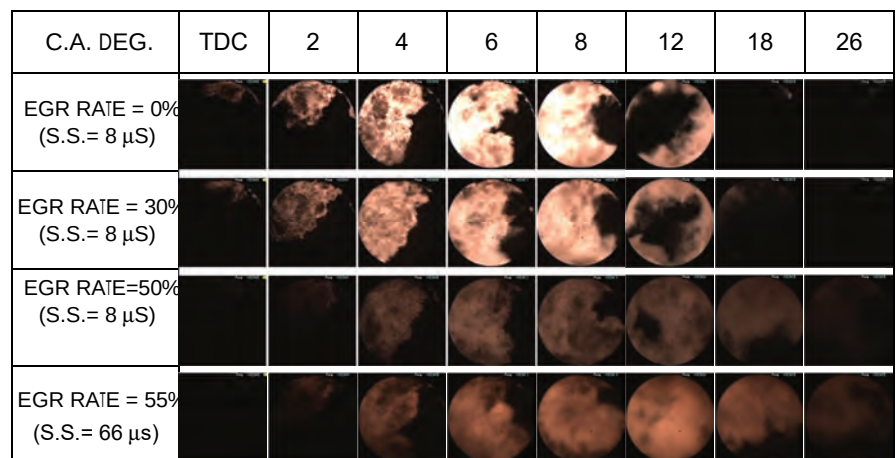


Fig. 12. High speed flame photos with borescope by changing EGR rate

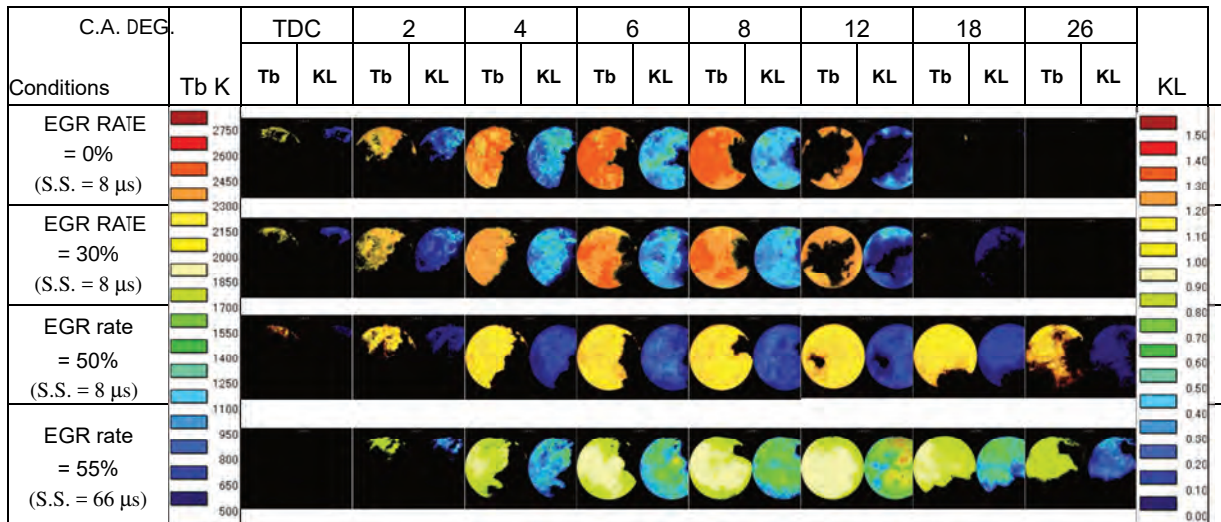


Fig. 13. Distribution images of flame temperature Tb and KL factor by changing EGR rates

The distribution of the flame temperature and KL factor to crank angle corresponding to these conditions are presented in Fig. 13, so that all EGR rate conditions from 0% to 50% can be compared. In the case of EGR rate 55% the flame temperature is low and KL factor is high in comparison with the case of EGR rate 50%. In the case of EGR rate 55% the smoke concentration (FSN) is high in exhaust pipe.

Figure 12 presents the sequential photographs of a diesel flame at each crank angle by the EGR rate increases. The flame brightness decreases significantly as the EGR rate increases, which implies that the flame temperature decreases as the EGR rate increases.

Figure 13 displays the analytical result of diesel flame at an enhanced EGR rate with the flame temperature software, flame temperature distribution, and KL factor distribution at each crank angle. The flame temperature decreases significantly as the EGR rate increases: 2500 K at EGR rate = 0%, and 1900 K at EGR rate = 55%. This flame temperature drop brings about drastic reduction of BSNO_x . The KL factor is also related to exhaust smoke because it increases at EGR rate = 55%.

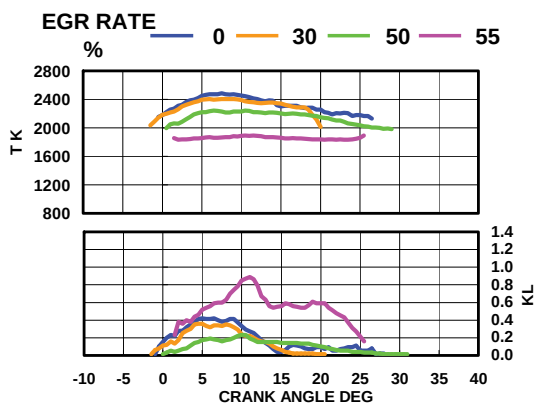


Fig. 14. Averaged Tb and KL factor by changing EGR rates

Figure 14 displays the variation of flame temperature and KL factor determined by flame images in the observation range of the borescope at each crank angle at EGR rate = 0, 30, 50, and 55%. The peak values of flame temperature

Tb vary largely from 2500 K to 1900 K by increasing EGR rate and the difference is large with 600 K. On the contrary the thermodynamic mean temperatures T_{cyl} in Fig. 11 vary a little from 1400 K to 1350 K and the difference is small with 50 K. The reason of BSNO_x reduction by increasing EGR rate can't be explained due to the decreasing of the thermodynamic mean temperature. Then it is very important to measure the flame temperature and the diesel combustion phenomenon in the cylinder should be understood for the reduction of exhaust emissions.

5.3. Increasing injection pressure

The change in ROHR as injection pressure P_{inj} is increased from 90 MPa to 150, 200 and 220 MPa is presented in Fig. 15 at EGR rate = 50%. Figure 16 displays the combustion photos at each crank angle from the ignition starting

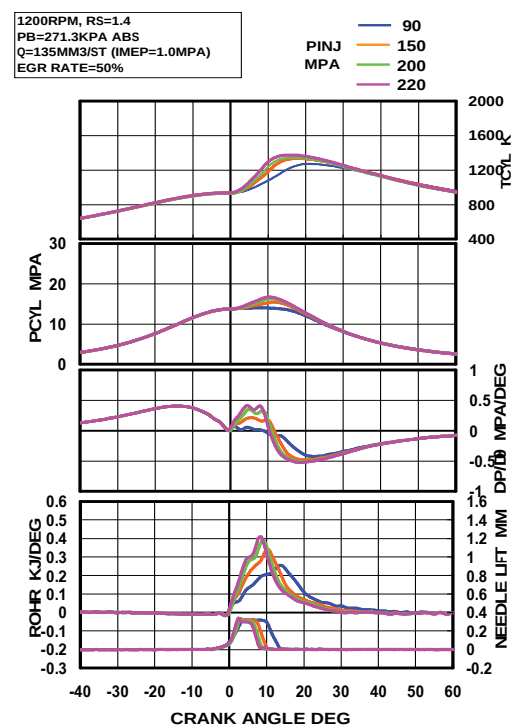
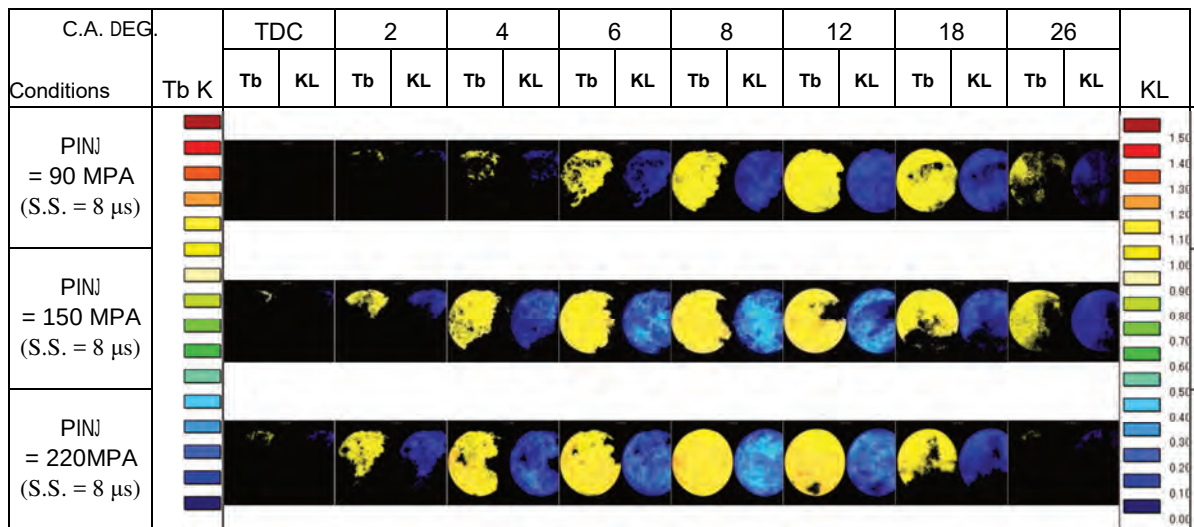
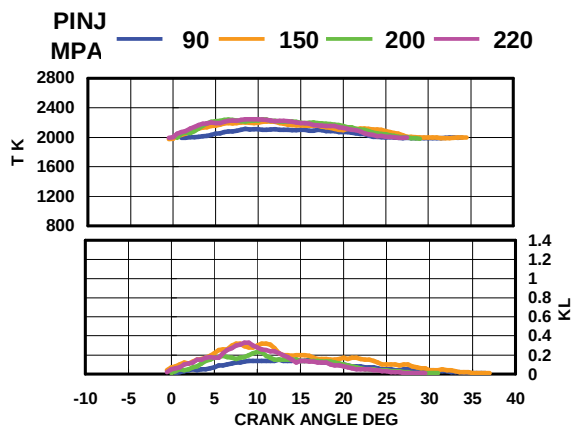


Fig. 15. ROHR by Changing Pinj at EGR rate 50%

C.A. DEG.	TDC	2	4	6	8	12	18	26
P _{inj} = 90MPa (S.S. = 8 μs)								
P _{inj} = 150MPa (S.S. = 8 μs)								
P _{inj} = 220MPa (S.S. = 8 μs)								

 Fig. 16. High speed flame photos with borescope by changing P_{inj} at EGR rate 50%

 Fig. 17. Distribution images of flame temperature Tb and KL factor by changing P_{inj} at EGR rate 50%

period of TDC, so that conditions can be compared when P_{inj} = 90, 150, 200 and 220 MPa at EGR rate = 50%. The distribution of the flame temperature and KL factor are presented in Fig. 17 in the same way. Figure 18 shows their variation in reference to the crank angle. The case of P_{inj} = 200 MPa is abbreviated in Fig. 16 and Fig. 17 because of that is already shown in Fig. 12 and 13.


 Fig. 18. Averaged Tb and KL factor by changing P_{inj} at EGR rate 50%

P_{inj} is changed from 90 MPa to 150, 200 and 220 MPa at EGR rate = 50%. Although BSNO_x is already low and

smoke is high at P_{inj} = 90 MPa, it is necessary to raise P_{inj} to 150 MPa to reduce smoke. BSFC can be reduced further by increasing P_{inj} to 220 MPa. It is, after all, a necessary technology to combine high EGR rate and high pressure injection for reduction in exhaust emissions and fuel consumption, as widely acknowledged.

The flame photographs in Fig. 16 suggest that the brightness is low and combustion is not active at P_{inj} = 90 MPa, although combustion is active and brightness is high at P_{inj} = 150 and 220 MPa. The flame temperature at each condition of 6–12 deg. CA shows a peak of 2230 K at P_{inj} = 220 MPa, which is not too high temperature because the EGR rate = 50%, so that BSNO_x stays low.

As the author mentioned above, it is very important to measure the flame temperature and KL factor, which is related with soot amount. By the measurement the diesel combustion phenomenon in the cylinder should be understood well for the reduction of exhaust emissions such as BSNO_x, smoke and PM in a multi-cylinder engine. The author has improved successfully BSFC and exhaust emissions of a 6-cylinder heavy duty diesel engine by using high EGR rates [26].

6. Conclusion

The measurements were conducted on conditions of engine speed Ne = 1200 rpm and fuel amount of stroke q = 135 mm³/st, with a middle load where high boost engines

are frequently used on road as BMEP = 1.0 MPa around. Fuel injection pressure Pinj and the EGR rate were varied within 90–220 MPa and 0–55%, respectively.

The following results were obtained.

- It was confirmed that the diesel combustion flame temperature is measurable using the two-color method from digital data obtained with an electronic high-speed digital camera. Moreover, it was verified that diesel flame observation by the borescope system is useful. It is very important to measure the flame temperature and KL factor for the reduction of exhaust emissions.
- When the EGR rate was varied from 0% to 55% at Pinj = 200–220 MPa, BSNO_x dropped to 1/40 and the flame

temperature also fell from 2500 K to 1900 K, which corresponded well with the BSNO_x drop.

- As the fuel injection pressure was increased from 90 MPa to 220 MPa at EGR rate = 50%, the flame temperature tended to increase slightly. But Smoke decreases very largely and PM in exhaust pipe drops largely.

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Nomenclature

BMEP: Brake mean effective pressure, MPa
 BSFC: Brake specific fuel consumption, g/kWh
 BSCO: CO weight per effective power and time, g/kWh
 BSHC: HC weight per effective power and time, g/kWh
 BSNO_x: NO_x weight per effective power and time, g/kWh
 EGR rate: EGR rate by [CO₂] at intake air, %
 IMEP: Indicated mean effective pressure, MPa
 Ne: Engine speed, rpm
 Pb: Boost pressure, kPa
 Pinj: Fuel injection pressure (at Common rail), MPa
 Pmax: Maximum or peak cylinder pressure, MPa
 Pcyl: Cylinder pressure, MPa

q: Fuel quantity per stroke, mm³/st
 ROHR: Rate of heat release, kJ/deg
 Rs: Swirl ratio, –
 Smoke: Smoke, FSN
 S.S.: Shatter speed, μs
 Tb: Flame temperature by soot, K
 Tcyl: Mean cylinder gas temperature, K
 Tin: Intake air temperature included EGR gas, C
 dP/dθ: Rate of cylinder pressure rise, kPa/deg
 λ: Excess air ratio, –
 λ eng: Engine excess air ratio considering whole [O₂], –

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Application of biogas to supply the high compression ratio engine

The study concerns the use of biogas as a fuel for supplying a modified self-ignition engine. As a result of the modifications made, the compression ratio was reduced and the engine was equipped with an ignition system and an electronically activated engine throttle. The changes have made it possible to burn biogas in a high compression ratio engine. The paper presents the results of research conducted on a low power cogeneration system with engine that drives an electrical machine cooperating with a 380/400 V network. The analysis includes, among other things, the possibility of producing electricity using biogas. The paper presents the influence of regulatory parameters such as the volume and composition of the supplied gas mixture and the degree of throttle opening on the obtained engine operation indicators and the driven electric machine. The tests were carried out in relation to the obtained ecological indicators depending on the concentration in the exhaust of such substances as: HC, CO, NO_x.

Key words: biogas, high compression ratio engine, emission, low power cogeneration system

1. Introduction

Combustion engines could be powered with different fuels. One of important fuel groups consists of gaseous fuels. There are fuels as generator gas with very low calorific value [5] from hydrogen [6] and fermentation gas (biogas) up to methane CH₄. The properties of selected gaseous fuels are listed in Table 1. It should be noted that methane, which is the main component of biogas and natural gas is characterized by a high octane number. That allows it to supply engines with a high compression ratio and combust without knocking. Biogas can be made from agricultural waste, which makes it possible to use the energy contained in it.

Table 1. Properties characteristic for chosen gaseous fuels [8, 13]

Gaseneous fuel	Density, kg·Nm ⁻³	Calorific value, MJ·Nm ⁻³	Calorific value of stechiom. mixture, MJ·m ⁻³	λ index for the lower flammable limit	MON
Methane	0.714	35.8	3.18	2.00	110
Propane	1.963	91	3.42	2.06	95
Butane	2.588	118	3.44	1.70	92
Natural gas	0.695	34.7	3.4	2.10	100–110
Coal gas	0.468	13	3.35	–	95
Generator gas	1.015	5.65	2.6	4.35	105
Illumination gas	0.614	17	3.25	2.50	90
Biogas 54% CH ₄	1.276	19.3	2.71	2.16	110
Propane – butane (50%/50%)	2.080	96.5	3.35	1.91	95
Hydrogen	0.089	10.8	3.03	10	130

There are many opportunities to use energy from biogas. Very often, in the case of stationary applications, the conversion of energy contained in the biogas fuel is used for generating electricity. However, in order to improve the efficiency of the entire system, ie. to avoid energy losses,

CHP cogeneration systems are built to simultaneously generate electricity and heat. Further improvement of energy indicators is possible with the use of trigeneration systems in which, in addition to electricity and heat production, biogas energy can be used during the sorption process (adsorption and absorption of cold). Among the various methods of using biogas energy, one can distinguish the use of gas combustion in a piston combustion engine. For this purpose, compression-ignition or spark-ignition engines are used. In the case of an engine fueled with diesel fuel, some modifications are required. It can be use dual fuel supply system (diesel and biogas) or only biogas. In the second case, it is often necessary to change the some construction properties of the engine in order to lower the compression ratio and adding equipment with an ignition system similar to those used in engines operating in the Otto cycle [4, 14]. In many cases, particularly relatively simple construction engines, the introduction of modifications does not lead to higher technical and economic expenses [14].

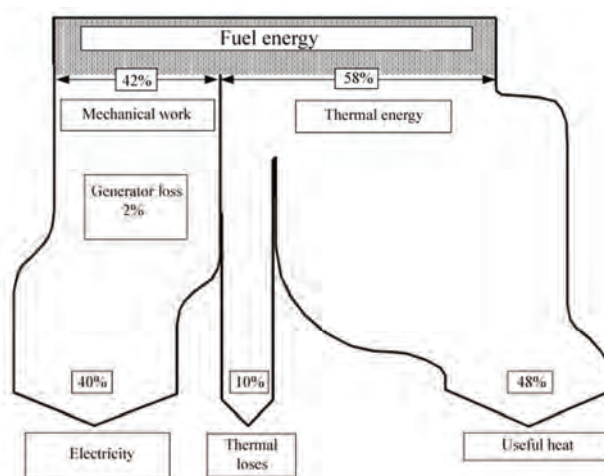


Fig. 1. Energy balance of an internal combustion engine operating in a cogeneration system [7]

Therefore, it is possible to adapt existing engine constructions for biogas power supply with relatively small modifications. In addition, the use of an internal combustion engine in a cogeneration system allows to achieve a high overall efficiency of up to 90% [7, 19]. Efficiency with the inclusion only electrical energy generation very rarely exceeds 40% – Fig. 1.

Other methods based on the use of, for example, Rankine, Stirling engine, micro-turbines or fuel cells are characterized by lower overall efficiency while working in a CHP.

2. Biogas and combustion engine operating indexes

The combustion of fuel in the engine is associated with the emission of harmful substances. Particularly in the diesel engine, it is important to reduce the emission of solid particles PM, but also the remaining other harmful exhaust components. In addition to the use of an environmental friendly design, the use of an appropriate fuel dosing system, and the aftertreatment method's, the type of fuel used and its detailed composition are important. Reduction of emissions is possible, for example, thanks to the use of catalytic fuel additives, which makes it possible to reduce PM emissions by up to over 60 percent [1]. However, the best results can be achieved using gas fuels, including those whose main component is methane.

Thanks to biogas supply, PM particulate emissions can be significantly reduce by up to 99%, carbon dioxide by 25%, and nitrogen oxides by up to 95%. At the same time, it is possible to achieve a reduction in fuel costs [20].

Both economic and ecological engine operating indexes depend significantly on the composition of biogas. Due to the fact that biogas can be produced using various technologies, as well as using different substrates, its composition and properties may differ significantly [2, 16]. Table 2 and 3 show the composition of biogas obtained using its various production technologies and the use of diverse substrates.

Table 2. The composition of biogas depending on the method of its production [2]

Komponent	Agricultural biogas	Biogas from energy crops	Landfill biogas
CH ₄ , %	45–75	57–62	37–67
CO ₂ , %	25–55	33–38	24–40
O ₂ , %	0.01–2.1	0–0.5	1–5
N ₂ , %	0.01–5.0	3.4–8.1	20–25
H ₂ S, ppm	10–30000	24–8000	15–400

Table 3. Substrates for the production of agricultural biogas and their composition [16]

Substrat	N ₂ , %	NH ₄ , %	P, %	CH ₄ , %
Maize silage	1.1–2	0.15–0.3	0.2–0.3	50–55
Grass silage	3.5–6.9	6.9–19.8	0.4–0.8	54–55
Rye silage	4.0	0.57	0.71	ca. 55
Pig slurry	6–18	3–17	0.2–1.0	60–70
Cattle slurry	2.6–6.7	1–4	0.5–3.3	60

One of the most important issues is the percentage share of CH₄ in the volume of biogas, which directly translates into such properties as the fuel calorific value. A change in the calorific value results in a change of the engine opera-

tion indicators, such as its power and torque. Due to the percentage volume of methane, the calorific value of biogas can vary within very wide limits of 15–27 MJ·Nm⁻³ [22]. Not without significance is the participation of other substances, including NH₄, which may have a negative effect on the engine components causing corrosion.

Research and analysis carried out by many researchers prove the desirability of further work related to the search for solutions that enable the use of energy contained in biogas during combustion in an internal combustion engine [10–12, 17]. Some researchers are focusing on dual-fuel solutions that do not require modifications to the CI engine design (including reduction of compression ratio, adding of the ignition system) [9, 17, 21, 23, 24]. As the results of the tests show, the method of controlling the dual fuel engine power is very important, taking into account the engine's operating status. The conducted research also shows the influence of, among other things, the degree of compression on the overall efficiency of the engine and its ecological properties [18].

Some research shows that the dual fuel supply system causes a decrease in the overall engine efficiency. Compared to the supply of only diesel fuel it decreases on average by approximately 17% (at full load as drop from 29% to 24%). At medium engine loads, the efficiency of the biogas supplied engine is approx. 20% in the case of unchanged compression ratio compared to the diesel supply. The higher compression ratio has a very favorable effect on emissions, resulting in a decrease of CO concentration by approx. 30%, HC by approx. 60%, but at the same time an increase in NO_x emissions by approx. 20% [18]. Also tests carried out for biogas supply show maximum values of the overall efficiency of the engine of 23.3% for a load of approx. 54%. This efficiency is slightly higher than with CNG 22.3%. In turn, with supply methane enriched biogas raises the maximum efficiency value to 26.2% [3]. Similar maximum values of overall engine efficiency were obtained in [15], slightly exceeding 22%.

Due to the required regulations regarding the use of renewable fuels, including Directive 2001/77/EC of the European Parliament and of the Council of Europe on "supporting the production of electricity from renewable sources on the internal market", they increase the share of energy from these sources. An additional decrease in the popularity of first generation bio-fuels (including agricultural crops) for the pressure to increase the second generation of renewable fuels (ie fuels created from products that cannot be used for human or animal food, i.e. most often different types of waste) shows that biogas can be seen as the fuel of the future.

As it results from official statistics presented by government institutions, the share of renewable energy sources in its overall production is still too small, and in particular it concerns the use of biogas in this agricultural biogas [25].

3. Research methodology

3.1. Test bench

In order to estimate the obtained ecological and economic indicators of the biogas-fueled research engine, a cycle of stationary tests was carried out on a test stand.

That consisting of an internal combustion engine prepared for operation in a cogeneration system that drives a three-phase low power electric machine connected to a 380/400 V network (Fig. 2). In this research program the possibility of heat recovery in the cogeneration system was not taken into account, but only the generation of electricity.



Fig. 2. Test bench

The test engine was modified (Table 4). It was equipped with an ignition system with the possibility of adjusting the ignition advance angle and an electronic control throttle mounted in the intake system and additionally with a high resolution speed sensor.

Table 4. Basic construction parameters of the engine

Parameter	Value	Unit
Mark	S-312B	—
Displacement	1.960	dm ³
Maximum power (Diesel)	20.6	kW
Engine speed for maximum power	2200	rpm
Maximum torque (Diesel)	100	Nm
Engine speed for maximum torque	1600–1800	rpm
Diameter/ stroke of the piston	102/120	mm
Compression ratio (original)	17.0	—
Compression ratio after modifications	15.0	—

Table 5. Basic construction parameters of the electric machine.

Parameter	Value	Unit
Mark	Indukta 2SIEL160M4	—
Power supply voltage	400	V
Rated power	11	kW
Rotational speed	1520	rpm
Current for rated power	21	A
Maximum torque	71.95	Nm
Efficiency with a load 2/4	88.2	%
Efficiency with a load 3/4	89.3	%
Efficiency with a load 4/4	89	%
Number of poles	4	—
Mass	105	kg
Inertia	0.061	kg·m ²

One of the important goals of modification was to obtain a lower compression ratio compared to the factory configuration of the S-312B engine, for which the compression ratio is $\epsilon_r = 17.0$.

As a result of increasing of the combustion chamber volume (realized by reducing the piston height), the compression ratio was reduced to $\epsilon = 15.0$.

A Indukta electric machine with the designation 2SIEL160M4 was connected to the crankshaft of the modified engine. Its basic technical parameters are shown in Table 5.

3.2. Fuel supply system

Due to the biogas composition variability occurring under real conditions, the cogeneration system engine was supplied by a system of gas fuel preparation with variable chemical composition. The biogas power supply system was collecting gases from CH₄ methane tanks (Linde – Methane 2.5 with a purity of more than 99.5%) and carbon dioxide CO₂ tanks connected by reducers R1, R2 and gas flow regulators M_{CH4}, M_{CO2}. Subsequently, the component streams of individual gases were directed to a tee located near the inlet connector of the mixer mounted in the engine inlet manifold (Fig. 3).

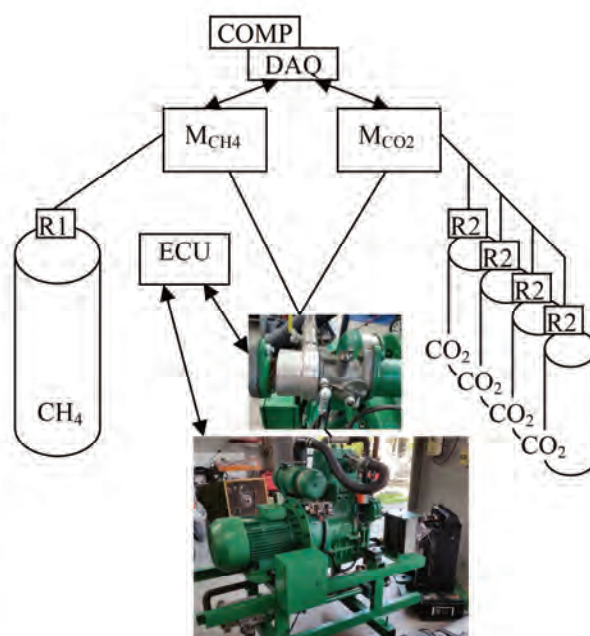


Fig. 3. Diagram of the engine's fuel supply system

The actual flow of the gas components supplied to the test engine was monitored by flow meters mounted on the supply lines of the relevant gas components. As the CH₄ methane flow regulator and flow meter, Mass-Stream mass flow regulator D-4371-DR was used to enable gas flow in the range of 0–130 dm³/min. Regulation and measurement of carbon dioxide CO₂ flow was achieved thanks to the Bronks mass regulator enabling gas flow in the range of 0–75 dm³/min.

3.3. Testing equipment

The research stand was also equipped with the Motorscan Leader 8000 flue gas analyzer in order to continuously record the gas concentration in the exhaust gases such as CO₂, CO, O₂, HC and NO_x.

The parameters of the current flowing between the electric machine and the network were monitored thanks to the

use of LEM type circuits. In turn, the amount of electricity produced is added up using the Orno OR-WE-505 meter with pulse output. In order to measure the current generated by the generator and the voltage in individual circuits of the 3-phase system, the HASS 50-S current transducer is included in the measuring path (Fig. 4).

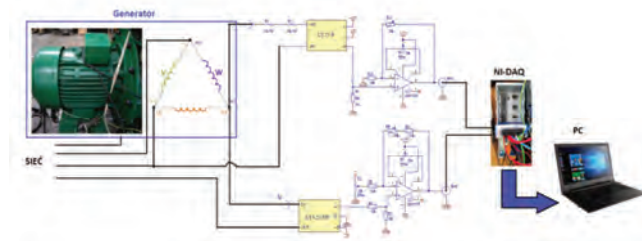


Fig. 4. Connection of the HASS 50-S current transducer with a test bench

The registration of all cogeneration system operation parameters (engine and electric machine) was possible thanks to the measurement system consisting of the National Instruments Compact DAQ module and the NI 9201 measuring card based on measurement system created in the LabView environment.

4. Results

The aim of the conducted research was to determine the impact of the throttle opening regulation and the composition of the gas mixture feeding the tested engine on system operation indicators such as electricity production and emission of harmful substances. The tests were carried out for different proportions of the $\text{CH}_4\text{-CO}_2$ gas mixture. Four types of gas mixture with the following CH_4 content were determined (in relation to the total flow thru the inlet of the mixer: 100%, 75%, 60%, 50%). So at example the mixture with volumetric participation of 75% of CH_4 and 25% of CO_2 will be further called 75/25. Simultaneously a gradual change of the throttle in the range of 10–100% was made (Fig. 5).

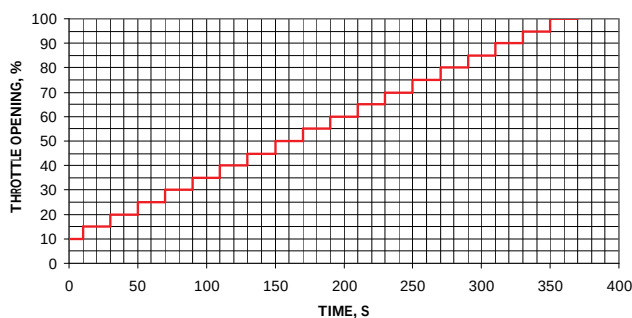


Fig. 5. The method of gradual throttle opening during the test

The results of tests and their analysis for fuelling the engine with gas mixtures with different CH_4 volume contents are presented below, however for a given CH_4 constant flow rate of $60 \text{ dm}^3 \cdot \text{min}^{-1}$.

4.1. Mechanical and electrical operation indicators

One of the important factors to assess the operation of a system consisting of an internal combustion engine and an electric machine is the generated electricity power and also

the rotational speed. The diagrams show the course components of electric power as the active, relative and apparent power. However, the most important is the comparison of active power values. Analyzing the time courses it can be stated that the value of active electric power in the initial phase increases rapidly together with the increase of throttle opening value. Exceeding, however, a certain value causes a sudden drop in electric power. To power the engine with a 100/0 gas mixture (pure methane), the highest values of active power (7.5 kW) occurring at the throttle opening of approx. 30–35% were obtained.

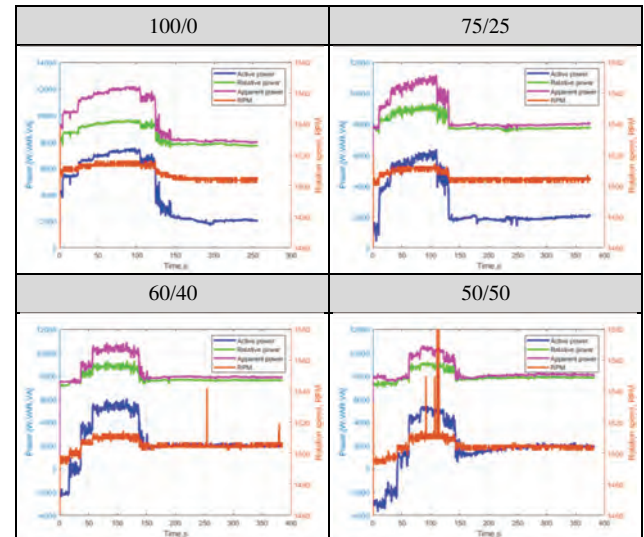


Fig. 6. Course of electrical powers and engine rotational speed at supply of different gas mixture

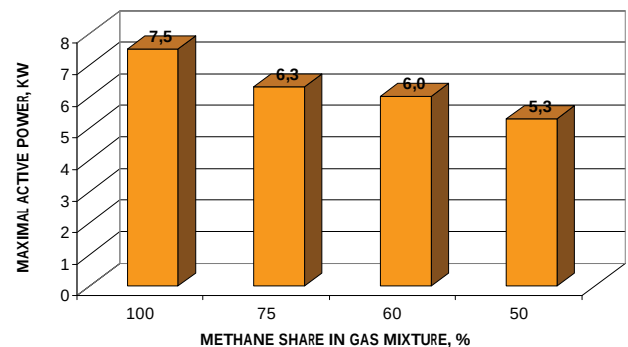


Fig. 7. Maximum value of active electrical power

For mixtures of gases with small share of methane, the maximum power values are respectively: for the 75/25 mixture 6.3 kW, for 60/40 6.0 kW and for the case of 50/50 fuel at 5.3 kW (Fig. 6, 7).

4.2. Ecological operation indicators

From the point of view of environmental impact, the emission of harmful substances should be monitored. The waveforms show the concentration patterns in the exhaust of individual harmful substances (Figs 8–10).

When analyzing the HC concentration in flue gas waveforms, it should be noted that the smallest values correspond to the throttle opening with obtaining the highest value of electric power (Fig. 8). However, supplying the

engine with a poorer methane mixture results in an increase in the HC concentration by several dozen percent (at the point of maximum electrical power of the system). In turn, the concentration of CO in the flue gas decreases with the increase of the throttle opening. Referring to the engine operation area associated with the maximum electrical power, it can be seen that it is close to the low CO concentration (Fig. 9).

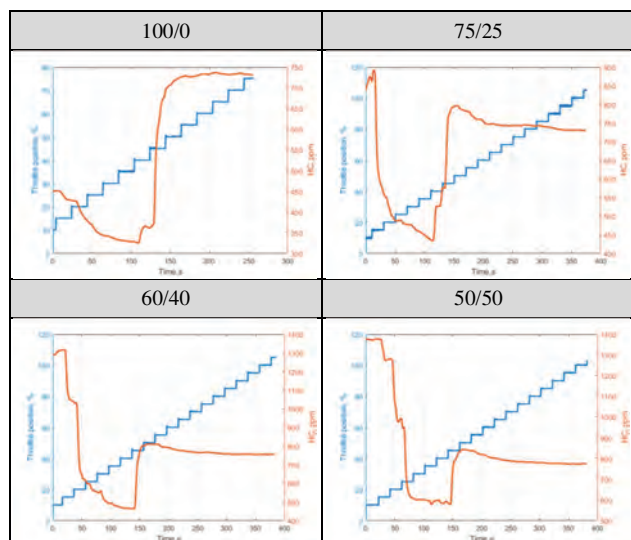


Fig. 8. Concentration of HC in the exhaust gases at supply of different gas mixture

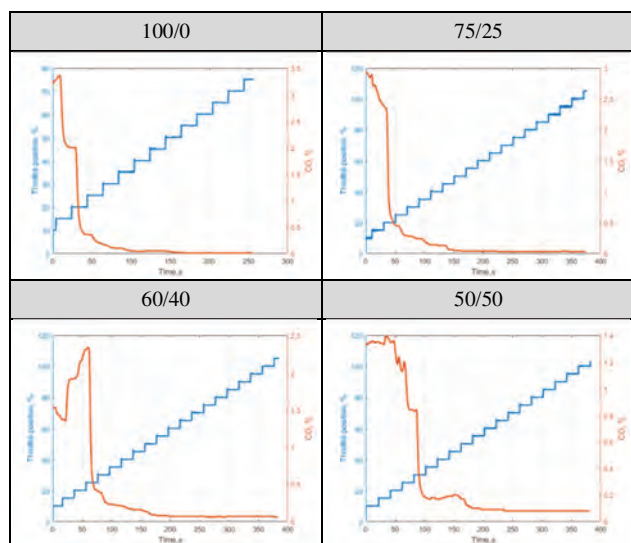


Fig. 9. Concentration of CO in the exhaust gases at supply of different gas mixture

Also in this case, the supply of a gas mixture with a lower volume fraction of methane may cause an increase in the concentration of CO in the exhaust. Analyzing the course of NO_x concentration, it can be seen an its increase in case of the high share of methane at supply mixture gases. In addition, high NO_x concentration values coincide with the throttle opening responsible for achieving maximum system efficiency (Fig. 10).

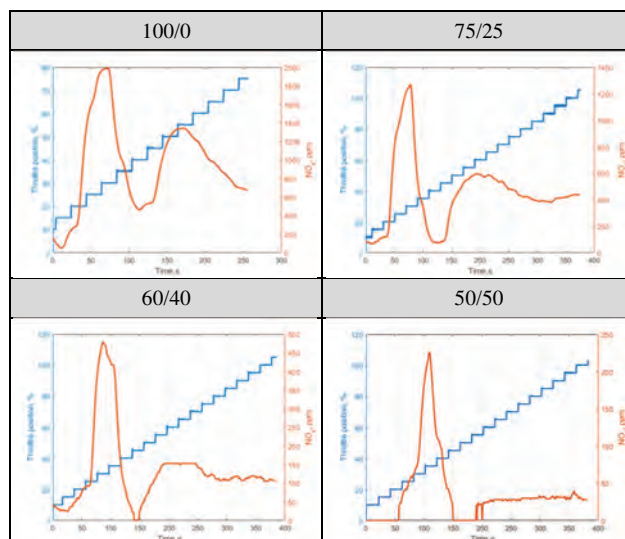


Fig. 10. Concentration of NO_x in the exhaust gases at supply of different gas mixture

NO_x concentration in the exhaust gas corresponding to the area of maximum efficiency is for the fuel supply with a high methane content (100/0) many times higher than for the case of its low volume fraction (50/50).

4.3. Fuel consumptions and overall efficiency

For many users of electrical energy systems, their economic properties are very important. For this reason, examples of specific fuel consumption courses are presented (Fig. 11).

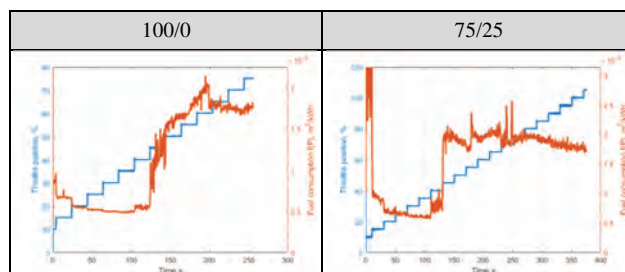


Fig. 11. Course of unit fuel consumptions

Its lowest values were recorded in the area of the highest electric power and they amount to $0.48 \text{ m}^3/\text{kWh}$ for 100 % methane content and exceed $0.6 \text{ m}^3/\text{kWh}$ for gas containing less than 60% methane (Fig. 12).

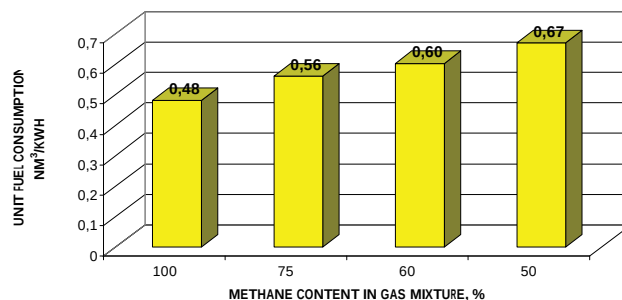


Fig. 12. Minimum unit fuel consumptions

The overall efficiency of the electricity generating system containing the internal combustion engine and the electric machine is determined as follows:

$$\eta_{eA} = \frac{P_A}{V_{CH_4} W_u} \cdot 100 \quad (1)$$

when calorific value of methane: $W_u = 35730 \text{ kJ} \cdot \text{Nm}^{-3}$.

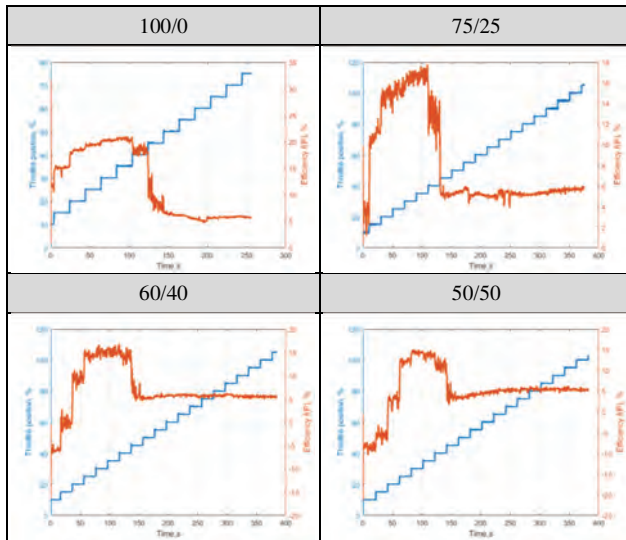


Fig. 13. Electricity generating system efficiency at supply of different gas mixture

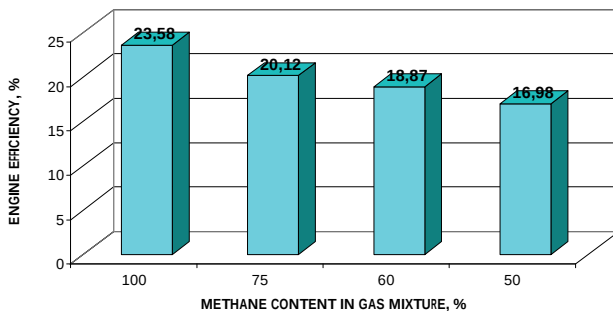


Fig. 14. Impact of throttle position on engine overall efficiency

Nomenclature

CHP	combined heat and power
CI	compression ignition
COMP	computer
DAQ	Data Acquisition Station
ECU	Electronic Control Unit
M_{CH_4}	CH_4 mass flow regulator
M_{CO_2}	CO_2 mass flow regulator
MON	motor octane number

As shown by the analysis of the overall efficiency of the engine-electric machine (Fig. 13), there is a clear range of throttle opening, where the highest efficiency value is obtained. It occurs for throttle up to 30–35%. Taking into account the efficiency of the electrical machine (Table 5), it is possible to estimate the overall efficiency of the internal combustion engine (Fig. 14).

Its highest values are obtained for the gas mixture with the highest methane content, amounting to 23.58%.

5. Conclusions

Use of biogas to supply a high compression engine can have measurable benefits both ecological and economical engine operating indexes. Obtained high efficiency is associated with the appropriate selection of the drive unit (matching the engine characteristics constant rotational speed of about 1500 rpm) but also with the appropriate control. As the research shows, in addition to the basic control elements such as the throttle and the regulation of supply gas stream, the obtained results are also significantly affected by the biogas composition. Low methane contents result in a decrease in the overall system efficiency and also in increased emissions of harmful substances. At the same time, the unit fuel consumption increases with respect to the drop of produced electricity. The most advantageous values of unit fuel consumption in the case of engine fuelling with low methane content (50%) biogas is $0.67 \text{ Nm}^3 \cdot \text{kWh}^{-1}$, and with high methane content (100%) allows to improve it by almost 30% ($0.48 \text{ Nm}^3 \cdot \text{kWh}^{-1}$). For this reason, one should strive to improve the methods allowing to increase the methane content in biogas, eg due to the reduction of the CO_2 in biogas volumetric concentration.

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e_f	compression ratio of Diesel engine
e	compression ratio after modifications
η_{eA}	overall electricity generating system efficiency, %
P_A	active power, kW
R1, R2	gas reductor
V_{CH_4}	methane consumption, $\text{Nm}^3 \cdot \text{s}^{-1}$
W_u	callorific value of methane, $\text{kJ} \cdot \text{Nm}^{-3}$

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Multiple symptomatic analysis of fails in the EGR system of HDI engine

The article presents the process of fault location in the HDI-rhr engine EGR system based on the characteristics obtained in the conditions of traction measurements using the ESI Tronic 2.0 program. The measurements were obtained using the licensed BOSCH KTS540 diagnostic tester. In the process of diagnostic inference the PWM signal coefficient has been analyzed in relation to the level of boost pressure, mass variability and air temperature in the intake system (based on MAF data) and EGR valve positioning for variable angular positions of the accelerator pedal. Analysis of the characteristics allowed for proper identification of faults previously stored in the system with OBDII error codes. Attention was paid to the issue of sediment formation in the EGR valve channels as a result of using diesel oil with the participation of biocomponents. Sludge analysis was carried out in a certified laboratory using infrared spectroscopy methods. Conclusions have been drawn for the presence of non-oxidized components of organic origin precipitated in the channels of the EGR system.

Key words: EGR exhaust gas recirculation valve, fluorescence, IR infrared spectroscopy, biocomponents

1. Introduction

The high degree of sophistication of the functional systems of the engine introduces the possibility of various types of damage and errors in the control system. Diagnostic procedures OBDII (On-Board Diagnostic level 2) allow locating faults and checking components in the state mounted in the vehicle, also providing data on the type of equipment fitted, software version and drivers used. Thanks to this, you can get data showing the cause of the failure of a given system in the car. In addition, diagnostic programs give the possibility of taking real-time measurements of selected parameters for such systems as, for example: HPCR, EGR, DPF. The OBD II system (EOBD) is a set of diagnostic tests and calculation, decision and real-time procedures. OBDII standards determine which systems in the vehicle are to be supervised, each controller demonstrating the implementation of at least one control procedure creating continuous or discontinuous monitors. The diagnostic monitor contains diagnostic procedures responsible for the control and observation of the components assigned to it. Engine manufacturers are required to publish an algorithm allowing to implement the driving cycle that guarantees the performance of all monitors provided for in the OBDII standard [9]. The ability of multi-symptoms inference and knowledge of technical solutions used in vehicle systems are the basis for shaping accurate diagnostic decisions. Identification of a multi-symptom engine fault with CS is usually a problem even for a diagnostician using dedicated testers. Detection of a fault in advanced vehicle structures requires from the diagnostician considerable knowledge in the scope of construction and functioning of mechatronic systems.

The EGR (Exhaust Gas Recirculation) exhaust gas recirculation system, which is often used in engines with low displacement, is often used to reduce the combustion speed in the explosive phase, thus contributing to the reduction of NO_x in the exhaust. However, an excessive share of exhaust gases in the entire charge delivered to the cylinder may cause a decrease in power and a significant increase in PM emissions, thus obtaining unfavorable values of other engine work indicators [2]. Limit value of the PM particulate

emission (Particulate Matter) is an equivalent criterion in relation to the emission of toxic exhaust components in SI engines [4]. These particles absorb heavy metals, sulfur and nitrogen compounds as well as polycyclic aromatic hydrocarbons, among which may be directly or indirectly carcinogenic substances [10].

2. Diagnostic technologies and procedures

The diagnostic testers used on the car services are terminals connected to the car's producer's servers, carrying out the diagnosis process in the Pass-Thru system. Professional diagnostic devices of such companies as Bosch, Hella-Gutmann, Texa, Delphi, Magnetti Marelli – also allow independent car services to connect to manufacturers' servers, download diagnostic and service data, update driver software, encode functions. Scanners of many companies are offered on the market, most of which carry out tests based on the ELM327 chip. The popular ones include scanners operating according to USB, Bluetooth and WiFi technology. USB scanners use PCMSCAN, Dash Command, ScanXL software with access to a Windows PC or Mac OSX and Linux. Scanner with Bluetooth technology works with Android, Symbian, Windows Phonem. It is also possible to use a PC with a Bluetooth receiver, but the collision takes place with a different iPhone protocol. Free applications from the Internet are available to read and delete errors, and control current parameters for selected drivers. Information is appearing more and more often about the possibility of taking over such applications to controlling selected functions of the vehicle. Scanners create their own WiFi network function with iPhones using application Dash Command, OBD Doctor, OBD Fusion as well as Android phones.

Car manufacturers use their own error determination terminology. Particular diagnostic testers can read errors in various ways, the coding of which does not comply with ISO standards. In this way, it may happen that the same error will be read in different ways by particular diagnostic testers [14]. The basic disadvantage of universal diagnostic testers is the inability to control all possible parameters, while the error coding is general and inaccurate, which

makes it difficult to indicate the fault. The advantage of the KTS series testers is the ESI Tronic software that provides extensive service information containing technical and repair data. High-tech testers enable to perform activation tests of executive elements, adaptation of controllers consisting in restoring factory settings, entering codes necessary to run executive elements. In addition, real-time parameter recorders of the motor operation are used as a function of time and are connected to the vehicle's diagnostic connector to diagnose occasional faults. It is possible to read and interpret the saved values in graphical form.

Despite advanced diagnostic technologies, determining the cause of the fault in the EGR system is problematic. The problem arises from determining the mass of recirculated gases as the difference in mass of the load and air mass in relation to the temperature correction of the intake air. In this case, the recognition of errors is a complex procedure, since a group of mutually coupled signals from the electric actuator of the system is analyzed, in relation to the mass flow signal, intake pressure, temperature factors and shaft speed [8]. The degree of EGR valve opening determines the exhaust mass for the purpose of ensuring the instantaneous value of the recirculation degree XEGR. The error interval is strictly set and the exceeding of this value determines the emission fault leading to the signaling of the engine operation status via the MIL indicator. The effects of these processes result in the occurrence of engine failures related to the record in the error code controller, the MIL indicator signaling, and switching to the engine's substitute operating characteristics.

The main problem is the inability to measure the specific heat of recirculated gases based on the specific heat of its individual components with the determination of their mass shares. There is also a variable heat exchange surface, differentiated heat transfer coefficient by the material structure of the intake manifold and a variable heat exchange time at the charge and ambient temperature difference. It is necessary to average the flow conditions of the factor with variable channel geometry in zones with different wall temperatures. Is important the lack of possibility to measure the mass flow rate of recirculated gases at variable temperature of the medium at the inlet to the cylinder.

The EGR system is distinguished by the degree of a significant tolerance range to momentary change in the recirculation rate, especially in the in the range of unsteady of transient engine loads. In addition, as a result of the use of biofuel additives, there is a differentiated, relative to hydrocarbon fuels, share in the exhaust of such components as: CO₂, CO, HC, NO_x [12].

3. Object of diagnostic tests

The object of diagnostic tests was engine the 2.0 HDI RHR with self-ignition, showing uneven idling. During the increase of the angle of the accelerator pedal, the engine went out, which indicated the emergency operation status, also signaled by the MIL indicator. Typical symptoms of uneven engine operation and power drop along with signaling of the MIL (Malfunction Indicator Light) indicator may have various causes [13]. Normally, the error codes indicate a fault in the fuel injection system, a fault in the

VTG turbocharging system or determine the condition of the EGR valve malfunction.

The car diagnostics process was performed in stationary conditions with the LEXIA 3 FULL CHIP diagnostic interface with the Peugeot Planet PP2000 software. The set works with the latest versions of Lexia, Proxia, Diagbox and Peugeot Planet programs. The stored code of the first error P0490 indicates a malfunction of the EGR due to incorrect positioning of the valve work piece in relation to the reference value. The next code P0401 referred to the erroneous values of flow air intensity as indicated by MAF. The main problem of identifying the damage resulted from the stored P1471 code during reading with the engine switched off, passing periodically in the transient error during the test on the running engine. The Diagbox intervention periodically eliminated the saved error code, which, however, returned again, especially after the dynamic acceleration of the vehicle as part of the traction test. Usually, the periodic failure may be related to the intervention made to remove sediments from the DPF system or VGT turbocharger or it results from their replacement of components without removing the previous data from the EDC controller [5]. Lack of software "adaptation" performed at a coolant temperature above 70°C may also cause transient errors. Periodically, the P0406 code is written which refers to the incorrect positioning of the electrically controlled EGR valve at too high control signal voltage.

In the first place, as a result of the test, correctness of basic work parameters was confirmed, including: the output pressure of the fuel pump (26 MPa at the crankshaft rev. 680 1/min) and the correct synchronization of the camshaft with crankshaft. The actual positioning of the EGR valve shows a small real difference (51.8%) in the ratio of the set point (52.1%) with a fill factor of 52%. However, there is a significant difference between the set value has a low value of the boost pressure (970 mbar) in relation to the required one (1010 mbar). The mass air flow value and the air temperature sensor data from the MAF flowmeter in the intake system were checked. At 12°C, the actual (455 mg) and required (610 mg) values differed excessively, which in the first place determines the defect of the flowmeter suggested by the saved error code P0401.

The initial stages of the engine diagnostic process carried out in stationary tests with indication of probable damages to both the EGR and MAF systems required extending the interpretation of the operating states of the engine's functional systems by additional records of the actual parameter values during the traction test.

As indicated in code P0401, replacing the MAF flowmeter would lead to a slight improvement in the exploitation of the engine. Therefore, it is necessary to aim for a full picture of the operating parameters of the interdependent systems and not individual symptoms or to replace the elements according to the error codes record [3].

Therefore, traction tests were performed with the actual engine parameters saved by the BOSCH KTS 540 diagnostic tester with the ESI (Tronic) 2.0 program. The actual values of the required and measured boost pressure, positioning of the EGR valve, engine speed and driving speed were obtained (Fig. 1). The record of error codes showed,

in addition to P0490, an additional code P0102 regarding the signal from the MAF flowmeter. The test clearly indicated a failure of the EGR valve as a result of incorrect positioning of its moving elements, which caused an excessive participation of exhaust gas recirculation and a reduction of the mass air flow rate. A list of saved error codes during the test under stationary conditions and traction test is given on the Table 1.

Table 1. Record of error codes during the test under stationary conditions and traction test

error code	description	Tester
P0490	Signal of electrical position of EGR valve: Positive loop deviation	PP2000 KTS 540
P0401	Fault in the air circuit in the power supply system	PP2000
P1471	Throttle control electromagnetic valve (EGR): open throttle	PP2000
P0102	Flowmeter signal; measured expenditure is smaller than the minimum expenditure	KTS 540

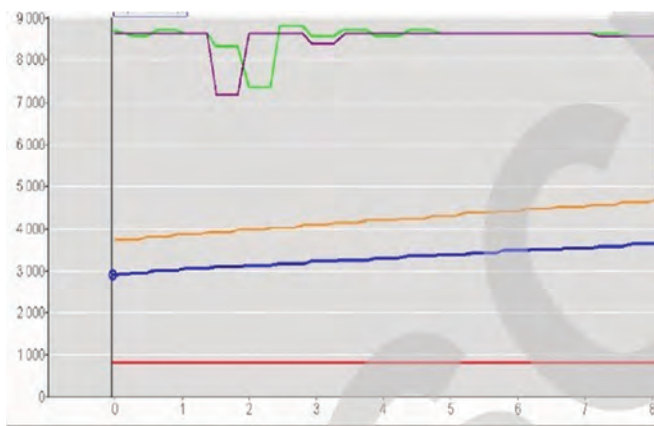


Fig. 1. Record of real parameters during the test as a function of time (s). Markings: boost pressure required – purple line, boost pressure measured – green line, travel speed – yellow line and engine speed – blue line (max. Test 102 km/h at 2300 1/min), red line – EGR valve positioning

Re-test traction was, after replacing the complete EGR valve, showed the correct flow characteristics of the actual boost pressure (green line) in relation to the required one (violet line). There is only a small shift (about 0.5 s), resulting from the inertia of the VGT turbocharger (for a normative pressure over 2300 mbar) when increasing the speed with the EGR valve closed confirmation of the accuracy of the diagnostic process is the recording of the actual parameters within the traction test phases (Fig. 2):

- acceleration phase with change of gear (ca. 20 s),
- phase of maintaining a constant speed of ride (min. 5 s),
- phase of releasing idling (ca. 20 s),
- engine braking phase (ca. 15 s).

In the acceleration phase of the car, the proper functioning of the EGR valve can be observed, indicating significant modulations of the exhaust gas recirculation rate. While maintaining a constant car speed, the exhaust gas recirculation rate is maintained at around 40–60%, for the idling phase and during engine braking it reaches 70–80%.

The main cause of the fault leading to the damage of the EGR valve turned out to be an excessive PM deposit in the zone of channels and mushroom valve in the EGR system.

Attempts at mechanical and chemical cleaning (hexane) of the stem and valve seat areas have not produced any results. Initial control of the basic engine parameters showed a good diagnosis. The last step is to delete errors saved in the controller's memory. This is an action that should always be done after repair or replacement of damaged elements [13].

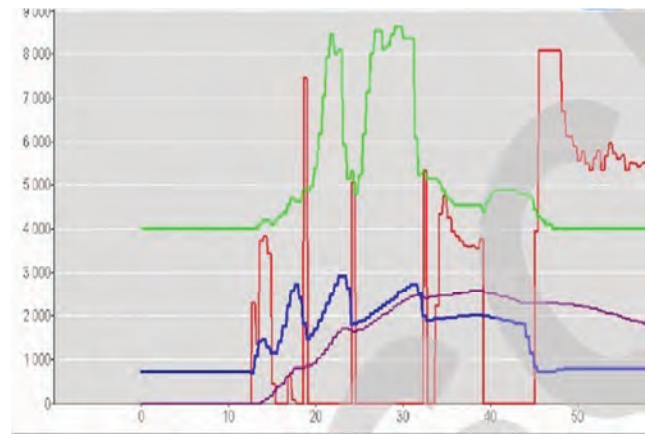


Fig. 2. Record of real parameters during traction test as a function of time (s). Markings: engine speed – blue line, fill factor for the exhaust gas recirculation valve (recirculation level) – red line, boost pressure – green line, car speed – purple line

4. Spectral analysis of sediments in the exhaust gas flow zones

The supplementation of diagnostic methods beyond the scope of EOBDII tests is the indication of various aspects expressing the causes of the failures. The inevitable consequence of fuel combustion in piston engines are IDID (Internal Diesel Injector Deposits) deposits created on work elements, causing a number of negative effects: deterioration of starting capacity, power drop, increased fuel consumption, increased emission of toxic exhaust components [6]. The functioning of HPCR, DPF, SCR and EGR systems in the SI engine depends to a large extent on meeting the fuel quality criteria included in the "Fuel Requirements for Diesel Fuel Injection Systems Diesel Fuel Injection Equipment Manufacturers Common Position Statement 2009" and the provisions contained in the World Card Fuels for category 4 diesel [7]. In the case of compression-ignition engines with direct fuel injection, the quality of fuel determines the speed of destruction of functional systems which is conditioned by the composition of fuel, fatty acid residue, a package of enriching additives and solid impurities. The type and physicochemical properties of the fuel (viscosity, density, fractional composition, cetane number, lubricity) and operating conditions of the self-ignition engine – significantly affect the mechanism of sediment formation. The results of many research works [1, 3, 6] indicate the complexity of the sediment formation process depending on the content of metal salts in diesel fuel containing FAME or ashless polymeric materials. Gas oils in the commercial distribution may contain various acid components (unsaturated fatty acids) with a lubricating additive character, which may favour the formation of sediments due to the acidic impurities contained therein and the presence of metal ions. Overlapping deposits in the

EGR valve duct zone are difficult to remove, as shown in Fig. 3, as a comparison of two valve seat images in the state of the saved error code and after partial cleaning with carbon dioxide removal agents. The macrostructure of the sediments in the microscopic image on the external surface of the mushroom EGR valve is presented on the Fig. 4. Similarly, hard-to-remove deposits and traces of seizure of the pin of the mushroom EGR valve were observed in the cooperation zone with the guide sleeve (Fig. 5). Improvement of the functional status of the EGR valve by washout sediments with active preparations leads to a partial and short-term improvement of its condition.



Fig. 3. Comparison of two pictures of EGR valve channels: channel with sediments (top picture), channel after partial removal of sediments with chemicals (bottom picture)

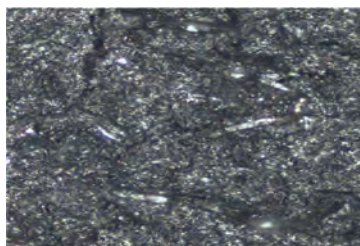


Fig. 4. Microscope image (magnification 100x, reflected light) of the sediment on the outside the mushroom of valve EGR



Fig. 5. Sediment on the bolt mushroom EGR valve in the co-operating zone with the guide sleeve.

On the bench for computer image analysis, a photo of the bolt mushroom EGR valve was made in the cooperation zone with the guide sleeve using a QImaging digital camera, Micro Publisher 5.0 RTV with a resolution of up to 2560×1920 pixels, with a linear polarizing filter. The Image Pro Plus 7.0 Media Cybernetics software was used to assessment of the percentage share of the area covered with sediment. After improving the quality of the obtained photo, the hi-pass filter (7×7) was transformed into black and white (image after quality improvement and black and white version) and then segmented (image after segmentation). Comparison of the share (black fields) in the area of the analysed sample showed the adhesion of sediments at the level of 61.4% of the bolt surface. In the sediment study process, spectral analyses were performed with a focus on identifying the chemical composition in order to determine the impact of the addition of biocomponents to diesel oil on

the analysed process. The application included the evaluation of the content of precipitated non-oxidized organic components. The spectral analysis of sediments taken from the flue gas flow channel and the zone of the mushroom valve seat was carried out [11]. The infrared fluorescence spectrum (FTIR) was recorded with the FTS 175 produced by BIO-RAD (Fig. 6).

The band in the diagnostic area of $2000\text{--}900\text{ cm}^{-1}$ is also intensive, which indicates the presence of oxidation processes of organic compounds to carbonyl and carboxylic structures. The effects of these compounds on nitrogen oxides are related to the presence of hydrated carboxylic acid salts. The effect of these compounds on the nitrogen oxides is related to the presence of hydrated salts of carboxylic acids. The 1650 cm^{-1} band is derived from other compounds containing C-O-NO₂ bonds, resulting from nitro-oxidation of engine oil components and fuel in contact with nitrogen oxides. The 1747 cm^{-1} band is visible due to the presence of degradation of FAME esters or di- and trimerization. For the purpose of identifying sediments of the ester group, the relations of carbonyl groups C = O (aliphatic) with the corresponding band in the area of the wavenumber $1750\text{--}1735\text{ cm}^{-1}$ are representative. A band of about 1720 cm^{-1} is derived from other carbonyl or carboxylic compounds, while a band of about 1633 cm^{-1} indicates the presence of products of interaction of nitrogen oxides with fuel oxidation products. The 1620 cm^{-1} band is derived from other compounds containing C-O-NO₂ bonds, resulting from nitro-oxidation of engine oil components and fuel in contact with nitrogen oxides. The weak 1735 cm^{-1} band is derived from other compounds containing C = O bonds resulting from the oxidation of engine oil and fuel components. The observed intense 1735 cm^{-1} band is related to the presence of esters, for which the representative ones are the relations of C = O (aliphatic) carbonyl groups with the bands in the $1750\text{--}1735\text{ cm}^{-1}$ range.

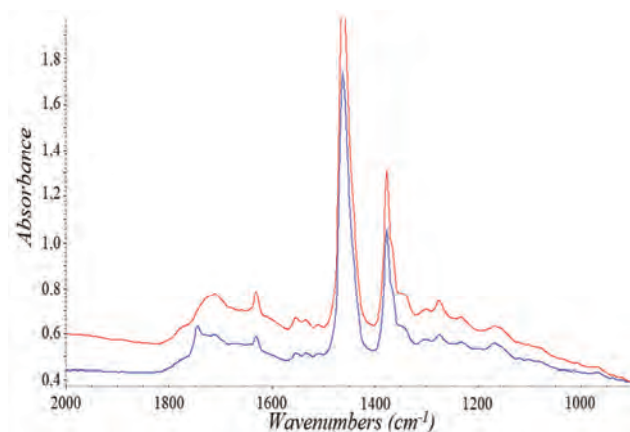


Fig. 6. Infrared spectrum (range $2000\text{ cm}^{-1}\text{--}900\text{ cm}^{-1}$) of a partial of soluble sediment taken from the flue gas duct (red) and the zone of the mushroom valve seat (blue)

The presence of ester structures are is also confirmed the bands in the area of "finger print" (ca. 1460 cm^{-1} and 1380 cm^{-1}) [11].

Reducing the amount of sediment on work elements the use of a more effective detergent-dispersing additive package, thus guaranteeing the desired performance. Infrared

spectroscopy analysis made after extraction of the sample showed clearly the proportion of organic compounds in the solid sludge. The main sources of sediment are the FAME polymer groups and components derived from various refining packages used for the production of biofuels.

5. Conclusions

1. The presented stages of testing and diagnostic inference in the case of multi-symptom states of motor failure along with a wide analysis of the recorded error codes, direct the procedure of inference to make the right diagnostic decision.
2. The main cause of the fault leading to the damage of the EGR valve turned out to be an excessive PM sediment in the zone of channels and mushroom valve in the EGR system.

3. Infrared spectroscopy analysis showed the proportion of organic compounds in the PM solid sediment. An analysis of fluorescence spectrum with the energy dispersion in relation to determining the proportion of sedimentary elements, together with the infrared recording of the spectroscopy spectrum showed the proportion of organic compounds in solid PM.
4. It is desirable to fully visualize the operating parameters of the interdependent systems and not to focus solely on the direct reading of the error codes indicated by the diagnostic tester.

Acknowledgements

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Nomenclature

CO	carbon monoxide	IDID	Internal Diesel Injector Deposits
CO ₂	carbon dioxide	MAF	Mass Air Flow
DOC	diesel oxidation catalyst	MIL	Malfunction Indicator Light
DPF	Diesel Particulate Filter	NO _x	nitrogen oxides
EDC	Electronic Diesel Control	PM	Particulate Matter
EGR	Exhaust Gas Recirculation	OBDII	On-Board Diagnostic level 2
FAME	Fatty Acid Methyl Esters	SCR	Selective Catalytic Reduction
HPCRS	High Pressure Common Rail System	VTG	Variable Turbine Geometry

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Coolant pump for compression-ignition aircraft engine

The article presents an analysis of the design of cooling liquid pumps for a compression-ignition aircraft engine. A 100 kW twin-charged, two-stroke, liquid-cooled engine has 3 cylinders and 6 opposed-pistons. In the first part of the study, the amount of heat needed to be removed by the cooling system was estimated to obtain the required volumetric flow rate. Then, the design of automotive cooling liquid pumps for compression-ignition engines with a Common Rail power supply system and power of about 100 kW was analyzed. The aim of the analysis was to select a suitable pump for applications in the aircraft compression-ignition engine. 5 constructions of different shape, diameter and width of the working rotor were selected. The pressure and volume flow rate were determined for a given rotational speed of the pump on a specially built stand. The operation maps of individual pumps were created to select the most efficient types of pumps.

Key words: diesel engine, opposed piston engine, aircraft propulsion system, cooling system, water pump

1. Introduction

The main source of heat in internal combustion engines is fuel supplied to the cylinder. The more fuel is supplied to the cylinder, the more heat is obtained from the cylinder as a result of combustion. Part of the heat passes through the walls, heating up the engine. In addition, a significant share is the heat generated by friction. The main source of friction is the crank-piston system, especially the friction between the sleeve and the piston. It reduces engine performance because of its deteriorated mechanical efficiency. At the point of friction, the temperature must not exceed a certain limit due to the thermal strength of the oil film. The engine oil temperature limit is the piston temperature measured at the bottom of the first piston ring duct at the level of 210°C [17]. Above this temperature, oil viscosity decreases and the tendency to lock the ring increases.

The main task of the Coolant Pump (CP) is to pump the coolant through the engine channels and the radiator. In most current cooling systems, the temperature of the coolant measured at the engine outlet oscillates between 80 and 90°C. The boiling point of the refrigerant in the cooling system must not be exceeded due to the possibility of steam plugs and the phenomenon of cavitation. In some engines operating under light and medium loads, in order to reduce fuel consumption, coolant temperature can be increased to about 110°C (temperature higher by about 15–20°C than the temperature considered suitable for the cooling system). On the other hand, in order to avoid damage to elements of the engine operating under high loads and for the engine to reach its maximum torque and power for the momentary values of rotational speed, it is necessary to quickly reduce coolant temperature by 5–10°C below the operating temperature of the cooling system of 90–95°C.

The authors of the paper [4] analyzed the CP parameters of 1.4 DV4 TED diesel engine from Citroen C3. The maximum temperature increase of the cooling liquid during its flow through the engine block is in the range of 10–15°C with the flow rate of the cooling liquid ranging 96–192 dm³/min. Mean CP efficiency is 20%. When the pump is running at low speeds with the thermostat closed, the total

efficiency is reduced to 15%. The main losses of the centrifugal pump are due to secondary flows and changes in the direction of speed after passing through the impeller. Hydraulic power can be lower than the power absorbed by bearings and conventional pump seals. The maximum lifting height of the pump, expressed by the pressure increase between the inlet and outlet at a closed thermostat, is up to 170 kPa [4].

Continuous suction centrifugal pumps are used in engine cooling systems. Their function is to transfer kinetic energy to liquid particles by means of blades of the rotating rotor. The liquid is subjected to a centrifugal force capable of increasing pressure and kinetic energies. The increase of liquid energy depends on the rotor design and its rotational speed.

The classic indirect cooling system of an internal combustion engine consists of a pump (usually centrifugal), which presses cooling liquid into a cooling jacket including channels in the engine block and the head. The liquid is then separated in the thermostat into a short circuit from where it returns to the pump or cooler.

The primary objective of the research on the modification of the cooling system is to reduce fuel consumption and thus CO₂. The efficiency of a classic cooling system can be increased by improving components such as the thermostat or pump [8, 18, 20] and by optimizing the cooling jacket by CFD (Computational Fluid Dynamics) calculations. Major modifications of the cooling system consist mainly in replacing the centrifugal pump SVRP (Sliding Vane Rotary Pump) which has a variable capacity or a coupling disconnecting the pump [19]. There are also used pumps with an electric drive [1, 11, 15]. Such a solution cannot reduce energy consumption over periods of low load. Another popular solution is the use of a regulated thermostat [2] in order to increase the temperature of the cooling liquid in periods of low load and decrease its temperature for heavy loads [13].

2. Aircraft compression ignition engine CP.

The CP is designed for the new construction of aircraft engines with compression-ignition and power of 100 kW.

The liquid-cooled engine has 3 cylinders and 6 opposed-pistons. The engine with a capacity of 1.5 dm³ is charged by means of a mechanical compressor.

In the planned aircraft diesel opposite piston engine (OPE), the pump will pump the cooling liquid through the ducts in the engine block and radiator. In the classic design of internal combustion piston engines, the direction of liquid flow is perpendicular to the axis of the individual cylinders. In the analyzed solution, the flow will occur along the axis of additionally finned cylinders (Fig. 1). The direction of flow will affect the resistance of the flowing liquid, the increase in pressure and thus decrease in flow rate. An additional pressure drop will be on the radiator.

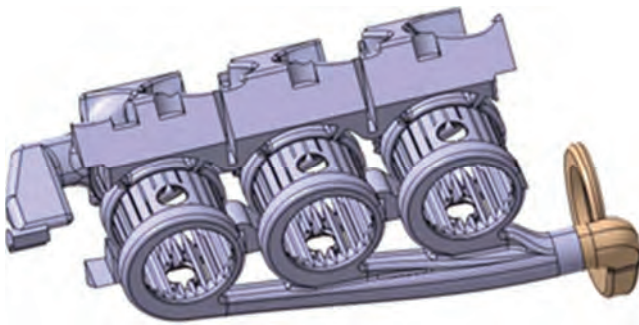


Fig. 1. Channels inside an OPE aircraft engine block with the flow along the axis of the cylinders

The car engine and its cooling system operate in unstable operating conditions. The aim is to shorten the warm-up time of the engine and adjust the pump capacity to the momentary load of the cooling system. Adjustable capacity cooling pumps are introduced. Alternatively, two-state control by disconnecting the drive or blocking the liquid flow in the pump housing. Additional electric pumps are also used to heat the car's interior.

The operation of the aircraft engine is stable. The heating phase takes place on the airport apron and is carried out according to instructions. Then, the engine goes to the operating state in which the aircraft takes off. In this phase, at the beginning, the cooling system absorbs the maximum heat flux from the engine. In addition, there is no airflow forced by aircraft motion. The crankshaft speed of the engine reaches its maximum value at the take-off power.

The CP mechanically driven by the OPE engine should have the lowest possible weight and low power requirement. These parameters will be influenced by the shape of the working rotor and the design of the bearings. The bearing shall be integrated in the drive roller. The main task of the CP is to pump the coolant through the engine channels and the radiator.

2.1. Volumetric flow rate requirements

Heat losses in the head of a classic internal combustion engine construction are significant due to the need to lubricate and cool its working metal elements. The construction of the engine with opposite pistons, whose the combustion chamber is formed by two piston heads and a sleeve, is free of the valve system and the head. The authors of the paper [12] carried out a thermodynamic analysis of an engine with opposed Achates OPS2 pistons. The heat generated is

proportional to the volume of the combustion chamber (V), while heat losses are proportional to its surface area (A). In an OPE engine, the A/V ratio is almost twice as small as in a classic engine. This results in higher thermal efficiency, which translates into a lower heat flux transmitted to the cooling system. Analysis of the OPE engine by means of the GT-Power software in the article [9] confirms that 59% of the maximum power of the 3-cylinder engine will be dispersed as heat in the liquid and oil cooling system, which in the case of the designed engine gives a value of 59 kW (59% of 100 kW).

The maximum volumetric flow rate of the pump to the designed aircraft engine was assumed on the basis of simulation tests in the Ansys Fluent at a level of 3 dm³/s (180 dm³/min), with a temperature difference of 5°C between the inlet (90°C) and outlet (95°C) from the engine block. The pump should provide a constant volumetric flow rate. Its power should be chosen in such a way that there is no significant drop in pressure associated with a reduction in the flow rate of the liquid flow.

In the AVL BOOST software, an OPE engine model has been developed to determine the energy balance of the motor, which is shown graphically in the Sankey diagram in Fig. 2 [10].

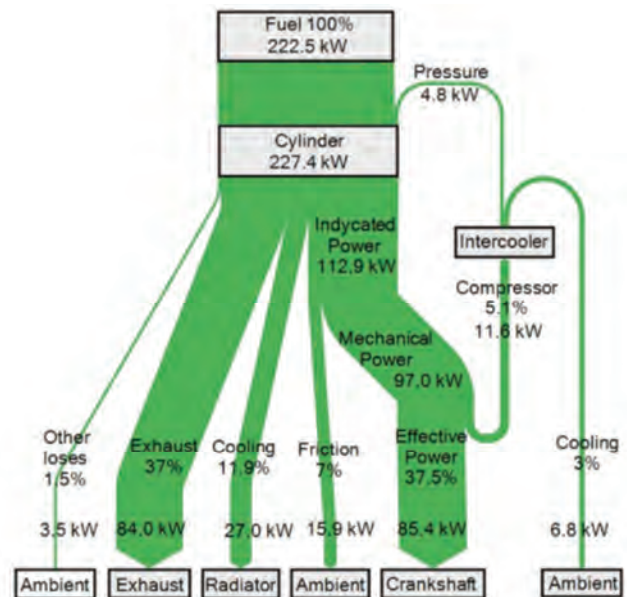


Fig. 2. Sankey's energy balance graph of OPEs aircraft engine with its mechanical compressor – Rotrex C30-64 [10]

It shows that the heat to be dissipated from the engine is the value resulting from the "Cooling" combustion chamber (27 kW) and friction (15.9 kW). The friction value also includes maximum power to drive the engine systems: common rail pump (3 kW), oil pump unit (0.8 kW), water pump (0.2 kW), alternator (0.6 kW). By subtracting the power per unit, the total heat output of the cooling system E (38.3 kW) is obtained. On this basis, the pump volumetric flow rate (VFR) can be calculated using the following equation:

$$Q = \frac{E}{C_p \cdot \Delta T} \quad (1)$$

where: Q – cooling liquid volumetric flow rate through the engine in dm^3/min , E – heat dissipated by the cooling system in kW, C_p – specific heat of the cooling liquid in $\text{J/kg}\cdot\text{K}$, ΔT – temperature difference of the cooling liquid between the output/input of the engine block (5°C).

When using water (specific heat as $4208 \text{ J/kg}\cdot\text{K}$ at 90°C) as a cooling medium, a volumetric flow rate of $109 \text{ dm}^3/\text{min}$ should be selected. When using Glysantin® G48 aero coolant mixed 50/50 with water (specific heat as $3412 \text{ J/kg}\cdot\text{K}$ at 90°C), the pump should have a VFR of $134 \text{ dm}^3/\text{min}$.

For comparison, the pump of the Austro Engine AE300 aero cooling system has a VFR of $140 \text{ dm}^3/\text{min}$. It is a 4-stroke 123 kW compression-ignition engine powered by the Common Rail system with 4 cylinders and a capacity of about 2 dm^3 .

2.2 Analysis of automotive coolant pumps

CP constructions of modern automotive compression-ignition engines with a power of about 100 kW were analyzed. The aim was to select automotive coolant pumps ideal for aircraft engines, to test them and to select the optimal design for the designed OPE engine. The following pump parameters were specified for particular types of vehicles: number of blades and an impeller diameter as well as the weight of the whole pump. The bolded bars in the diagrams (Figs 3–5) were selected for our bench tests. The speed ratio of the CP pump/crankshaft is 1:1. CP pumps can have a cast iron or aluminium housing (split or a uniform). The split housing has its drive part together with the rotor and cooperates with a water space formed in the engine block. The water space can be formed entirely in the block or partially formed in the pump body. The CP pump can be a separate or integral part of the engine. Such solutions are used in some Ford engine models and in most Japanese designs. The disadvantage of this solution is the large weight of the structure.

Figure 3 shows the rotor diameters of CP pumps in different car engines. According to the analysis, the diameter of the impeller of the CP pump for a diesel engine of 100 kW has a value in the range of 60–70 mm. The average value is 68 mm.

Figure 4 shows the number of blades of each pump. The pumps have an average of 8 blades located on the impeller. Their shape is straight or curved in one or more planes. The impellers of Citroen and Mercedes pumps are relatively low, so they have 20 and 12 blades at the perimeter, respectively. For comparison, the Mazda rotor is 18 mm tall and the Citroën – 9 mm.

Figure 5 shows the weight of CP pumps in each vehicle. The average weight of the pumps is 0.93 kg. The lightest CP pump is the one installed in VW/Skoda cars. The weight is reduced by mounting the bearing and sealing in the drive wheel. Additionally, the working chamber is completely in the engine block. Japanese pumps (Honda, Mazda, Toyota) and Ford cars are heaviest because they are made as separate elements mounted in different engines. These constructions have their working chambers in which the rotor works. The bearing with seals is mounted in the casing of the working chamber.

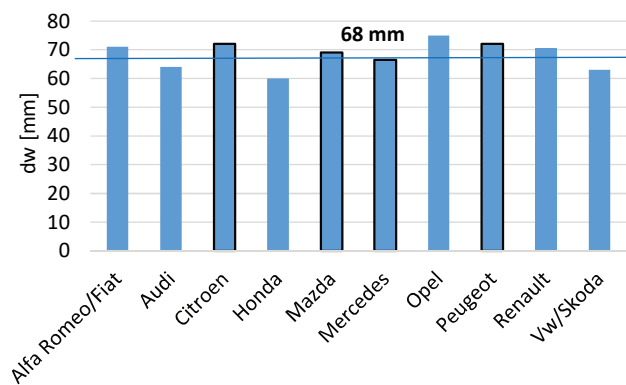


Fig. 3. Impeller diameters of each CP

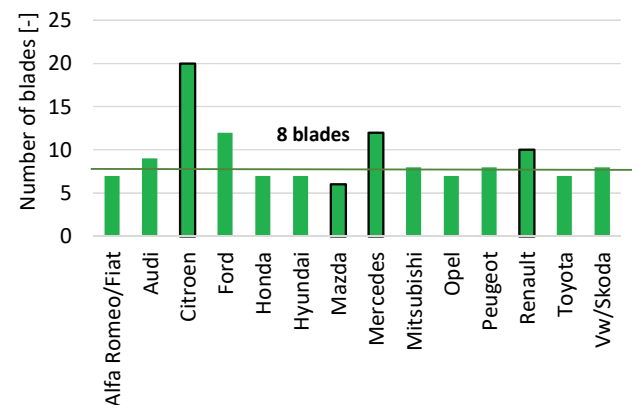


Fig. 4. Number of blades in each CP

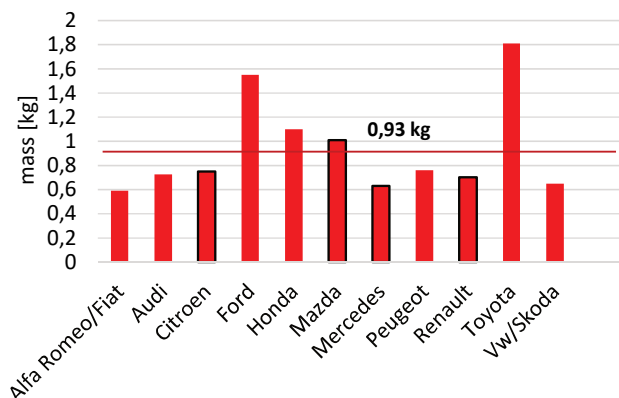


Fig. 5. Weight of each CP

3. Test stand

Figure 6 shows a diagram of the CP pump test stand. The Infineon KTY 19-6M/Z temperature sensor together with the Wika pressure tensiometer sensor has been installed in the operating chamber of each CP pump. To measure the volumetric flow rate, the BadgerMeter Blancett 1100 turbine flow meter measuring the flow rate from 19 to $190 \text{ dm}^3/\text{min}$ with an accuracy of 1% was used. To measure the pump rotational speed, a magneto-inductive sensor – Honeywell 1GT101 was used. A $1\frac{1}{4}$ inch volumetric flow control valve was mounted at the outlet from the pump chamber. The internal diameter of the valve is equal to the largest diameter of the outlet channel from among the

pumps selected for our testing (50 mm). On the stand, an electric motor of 3 kW power was installed, selected on the basis of the maximum power of the liquid stream – the product of the maximum output of 180 dm³/min and the pressure increase of 170 kPa [4].

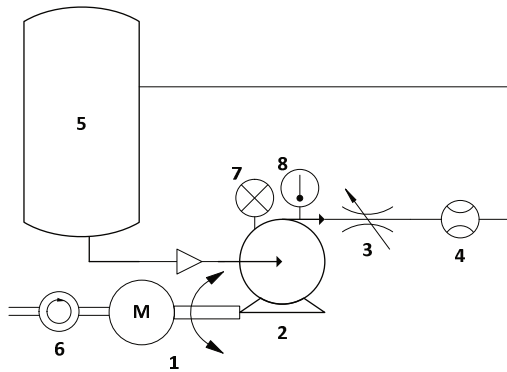


Fig. 6. CP test stand diagram: 1 – electric motor with an inverter, 2 – pump, 3 – throttling valve, 4 – flow meter, 5 – tank, 6 – RPM sensor, 7 – pressure sensor, 8 – temperature sensor

Four types of pumps were selected, differing in the rotor design and the shape of the working chamber. Table 1 presents the parameters of the tested CP.

Pumps for our testing were selected in line with the following criteria:

1. the smallest working chamber of the pump in the engine block
2. the smallest possible bearing and so clearance of the drive roller from the working rotor
3. the shape of the pump rotor – during the tests, the pumping efficiency of individual rotor designs will be determined.

Table 1. Parameters of analyzed CP [21]

Manufacturer	mass [kg]	pump rotor	
		diameter/height of the working rotor [mm]	number of blades
Mazda	1.01	69/18	6
Citroen	0.75	72/9	20
Mercedes	0.63	66.5/11.5	12
Renault	0.702	70.5/10	10

 Mazda	 Citroen
 Mercedes	 Renault

The reverse-engineering technique reproduces the working chamber of each pump that is located in the engine block. The holes and pockets of the pump impeller and water channels are covered with a thin foil. They were then filled tightly with a plastic material (plasticine) which also

covered the surfaces of the block to which the pump seal adheres. The model of the pump working chamber was scanned with a 3D scanner. The mapped geometry was imported into the Catia software which reproduced the internal geometry and designed the housing with holes for the pressure sensor, temperature and inlet and outlet channels. Figure 7 shows a schematic process of mapping the geometry of the pump internal chamber.

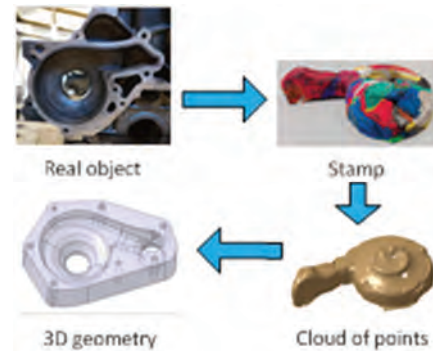


Fig. 7. Diagram of the process of mapping the geometry of the working chamber of the CP Citroen pump

4. Test conditions

For the pump speed n [rpm] set on the engine inverter, the valve position has been changed using the throttle. The following parameters were read: the volumetric flow rate Q [dm³/min] of the pump and the increase of the total pressure Δp_1 in the system, expressed as:

$$\Delta p_1 = p_1 - p_s \quad (2)$$

Before the measurements started, the value of static pressure p_s was read which is a result of atmospheric pressure and the height of the water column above the sensor. The tests were carried out for the initial water temperature of 25°C. The rotational speed of pumps was regulated from 1200 rpm to 7200 rpm every 300 rpm. The increasing rotation of the pump rotor increases both the flow and pressure of the liquid, which is a measure of the ability to overcome the resistances occurring in the cooling system. Flow losses occur in straight sections as well as in bends and constrictions when changing diameters. They are also related to the roughness of the walls in the block and in the radiator.

The products of pressure rise Δp_1 and mass flow rates Q determine the stream power expressed in Watts:

$$P_s = Q \cdot \Delta p_1 \quad (2)$$

The power of the stream P_s defines the work of the flowing liquid per a unit of time. The greater the work is done (the greater the pressure increase at constant flow rate or vice versa) in a unit of time.

5. Test results

Figure 8 shows the operating area of the individual pumps. The lower part is the flow characteristic without damping (Fig. 9) at different speeds. The upper part is a characteristic with damping for the rotational speed in the range of 6800–7030 rpm. Each point was obtained by changing the position of the throttling valve. The pressure drops as the volume flow rate increases. The flattest charac-

teristics is that of the Mercedes pump – the pressure does not drop despite the increase in the volumetric flow rate. Renault and Mercedes pumps have comparable shapes of characteristics. The Citroen pump shows a rapid drop in pressure with an increase in the volumetric flow rate.

All pumps (Fig. 8) show the same pressure drop as the volumetric flow rate increases from approximately 1200 rpm to 7000 rpm. The points with the highest flow rate and pressure were obtained at a rotational speed in the range of 6800–7030 rpm. The Citroen pump has a maximum flow rate of 82 dm³/min, Mercedes 172 dm³/min, Renault 150 dm³/min and Mazda 164 dm³/min.

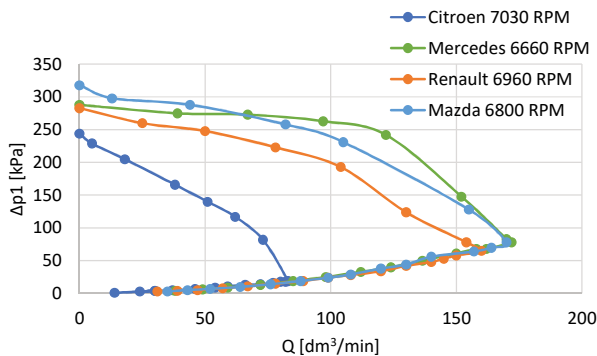


Fig. 8. Working areas of individual CPs – pressure as a function of volumetric flow rate

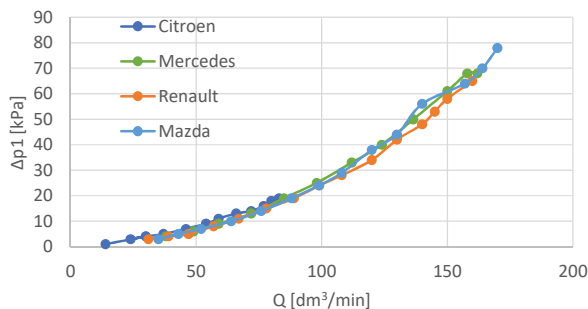


Fig. 9. Characteristics of individual CP pumps without damping – pressure as a function of volumetric flow rate

Figure 10 shows the full characteristics of the Mercedes Class B 180 CDI pump. The individual curves show the pressure generated by the pump as a function of the flow, expressed in liters per minute. The pressure is constant to approximately 100 dm³/min. As the volumetric flow rate increases (after exceeding 100 dm³/min), it starts to decrease. The decrease is larger for higher rotational speeds of the pump impeller.

Figure 11 shows the power of each CP stream as a function of the rotor speed without damping. It is very clear that the Mercedes pump has the highest stream power. At almost any rotational speed, the products of pressure rise and volumetric flow rate is greater than in the competitive designs. This means that the volumetric flow rate increases with the rotor speed as in all centrifugal pumps.

6. Summary

The analysis of the liquid cooling system shows that the following volumetric flow rates are required: 180 dm³/min (Ansys Fluent), 123 dm³/min (AVL BOOST), 108 dm³/min

(Austro Engine), given the use of water in the cooling system and a temperature difference of 5°C. If anti-freezing additives are replaced, the efficiency of the refrigerant is reduced, which means an increase in the required flow rate.

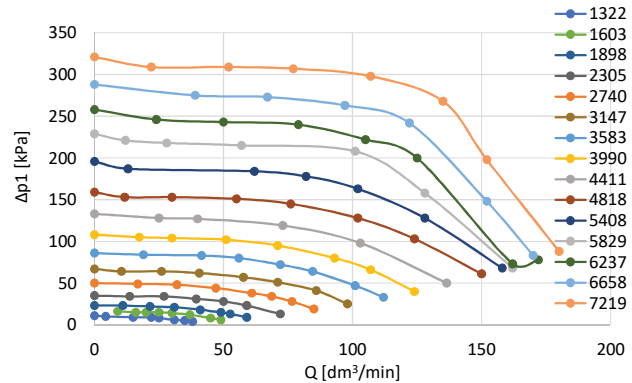


Fig. 10. Mercedes-Class B 180 CDI (damping at each speed) at individual speeds – pressure as a function of volumetric flow rate

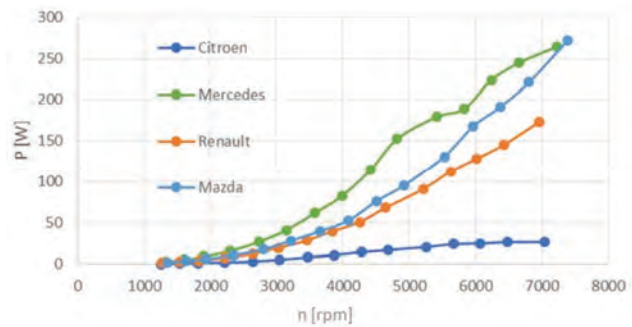


Fig. 11. Stream power of individual CPs as a function of rotor speed without damping

Most of the designs are blade impeller pumps in which part of the working chamber is in the engine block. Bearings and seals in the drive wheel must be used to reduce the weight of the CP pump. Additional weight reduction can be achieved by making the entire operating chamber in the engine block and using the polymer impeller and the aluminium pump casing. The analysis shows that the 100 kW diesel pump impeller should have a diameter of approximately 68 mm and 9 blades. The drive is transmitted by means of a belt ridge or a toothed belt that drives the camshaft.

Among the four cooling system pumps tested, the Mercedes pump installed in the B 180 CDI class has the highest performance. At lower rpm, compared to the competitive designs, it achieves a maximum flow rate of 162 dm³/min. Additionally, the pressure does not drop when the flow rate increases as in the other tested constructions. The pump has the highest flow power. Moreover, it is the lightest design (0.63 kg).

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Nomenclature

CP coolant pump
 OPE Diesel Opposite Piston Engine
 ICE internal combustion engine

CI compression ignition
 VFR volumetric flow rate

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Influence of electric motors assembly in hubs of vehicle wheels on the dynamics of movement, especially on surfaces with different adhesion coefficient

The article presents issues related to the drivability and stability of a vehicle equipped with electric motors placed in the hubs of wheels. In the article was presented the necessary modifications to the vehicle and their impact on the vehicle's motion parameters. Describes the behavior of the vehicle on a homogeneous surface and on surfaces with different adhesion coefficient under the wheels of the vehicle sides. The simulation model is presented. The last part of the article presents simulation results for compared surfaces with different adhesion under the wheels of the left and right side of the vehicle.

Key words: vehicle motion dynamics, electric vehicle drive, electric motors in hubs of wheels

1. Introduction

The simplicity of construction and control, high efficiency, no exhaust emissions, low noise and lower failure rate of electric vehicles contribute to increasing their quantity.

On the other hand the charging time, the location of the charging station and the weight of the batteries limit the development of electromobility. Intensive development works related with increasing the electric capacity of batteries and shortening their charging time allow to hope that electric vehicles will be widespread in the near future [7, 8].



Fig. 1. The in-wheel motors vehicle developed by Lohner and Porsche in 1898 [10]

Electric vehicles with motors built in wheels are increasingly being considered. This placing of the motors increases the inertia of the wheels and the unsprung masses affecting the dynamics of the vehicle motion, the steerability, the safety and the users comfort. The study of the unsprung masses impact on the dynamics of vehicle motion was taken by many authors [2–6]. The requirements for motors which must have a high torque to start and accelerate the vehicle are also increasing. The increasing of unsprung masses will also affect the behavior of the vehicle during braking, especially on surfaces with non-uniform coefficient of adhesion (μ -split). This issue is discussed later in the article. The article presents the results of braking and acceleration analysis of the vehicle on homogeneous surfaces and with different coefficients of adhesion under

the wheels of the right and left side. A method of assessing the behavior of the vehicle while moving on such surfaces is also proposed.

2. The vehicle modifications resulting from the installation of electric motors in wheels

In a car with in-wheels motors, there will be no power unit in the engine compartment; however, this place will be used for the installation of the controller, inverter, cooling system and battery. Suspensions and steering system must also be available there. It was assumed that the vehicle body weight will not change. The changes will apply to the wheels in which electric motors will be mounted. This assumption results from the fact that the anticipated electric vehicle will be built on the basis of an existing vehicle powered by an internal combustion engine.

The parameters of the SMzs200S32 electric traction motors built by KOMEL, intended for installation in wheel hubs, were used for the analyses.

These are synchronous motors excited with permanent magnets with liquid cooling (Fig. 2) [1].

The parameters of the SMzs200S32 electric motor are shown in Table 1.



Fig. 2. The 3D model of the SMzs200S32 motor made by KOMEL [1]

Table 1. Parameters of electric motor drive [1]

Parameter	Symbol	Value
Voltage	U_{LL} [V]	200
Nominal power	P_n [kW]	42
Nominal torque	M_n [Nm]	400 (for $n_n = 0-1000$ 1/min)
Max. power	P_{max} [kW]	~ 80
Max torque	M_{max} [Nm]	~ 90 (for $n_n = 0-850$ 1/min)

Motors with similar parameters are offered by Protean (Fig. 3) [9].



Fig. 3. Electric motor mounted in wheel hub made by Protean [9]

Based on preliminary analyses, it has been established that a B class vehicle ($d_{mc} \approx 1300$ kg), equipped with two motors of this type should provide the required driving dynamics. For cars with a heavier weight, a solution with four drive motors should be used to obtain a suitably high driving performance.

A motor can be divided into fixed elements and movable elements. The motor rotor is an external element consisting of a magnetic core with mounted permanent magnets. The non-rotating motor unit mounted to the suspension consists of an anchor disc, a labyrinth cooling system and a stator consisting of a wound magnetic core. The brake was built inside the wheel.

To analyze vehicle motion dynamics, the parameters of the SMzs200S32 engine were used. An alternative motor location was considered: in the rear wheel hubs, front wheels or in all class B wheels.

3. The scope of simulation tests

The aim of the authors was to show the differences in the behavior of vehicles powered by an internal combustion engine and electric engines placed in wheels. It was assumed that the mass of a vehicle equipped with an internal combustion engine will be similar to a vehicle equipped with an electric drive (only the masses of the wheels and their moments of inertia will increase). In the case of comparing vehicles with motors placed on the front or rear axle, all drive torque will be generated by these engines. In the event that both axes are driven, the torque will be separated by 50:50. The comparisons concerned the parameters of vehicle movement during a straight maneuver of acceleration and braking on the roadway of a high coefficient of adhesion and on a surface with different coefficients of adhesion under the wheels of the right and left side of the vehicle.

The following criteria were used to assess the vehicle motion for the straight line track: braking distance to stop the vehicle, deviation from the rectilinear direction of travel and the rate of braking intensity.

Simulation tests were carried out according to the established program.

A range of tests was adopted, in which the vehicle without drive motors placed in wheels was compared with a vehicle equipped with engines. The following cases of power transmission solutions with electric motors placed in the wheels were considered:

- engines placed in the rear wheels of the vehicle,
- engines placed in the front wheels of the vehicle,
- engines placed in all wheels of the vehicle.

The vehicle behaviour was analysed on the surfaces with:

- uniform coefficient of adhesion ($\mu = 0.8$) for attempts to accelerate and stop the vehicle,
- different coefficients of adhesion under the right and left wheels of the vehicle (μ -split, $\mu_1 = 0.8$, $\mu_2 = 0.3$) for attempts to accelerate and brake the vehicle.

The braking on the straight section of the road has been simulated, taking into account the operation of the ABS system. During the analyses, the own simulation program of the vehicle motion model was used [14].

4. A simulation model used for vehicle traffic analysis

In order to conduct a theoretical analysis of the impact of mounting of electric motors in vehicle wheels on the dynamics of motion, a mathematical model of the vehicle was developed.

In the model, the vehicle's basic movement was described using Cartesian coordinate system (position and orientation). The vehicle was treated as a set of rigid solids connected by flexible elements, resilient and damping. Flexible ties were imposed on the suspension movements and the turning movements of the front wheels.

To analyse the dynamics of the car a quasi-spatial model, which has 11 degrees of freedom, has been adopted. It takes into account:

- displacement, velocity and acceleration of the vehicle determined by the coordinates x_c , y_c (displacement of the vehicle mass center) and angle ψ of the vehicle rotation about z_c axis,
- angles of wheel rotation φ_k ($k = 1, 2, 3, 4$),
- a steering model that includes: the steering angle of the front wheels δ_1 , δ_2 , the rack movement x_L and the rotation angle of the steering wheel φ_w ,
- vertical movement of the vehicle z_c and rotation angles θ , φ (longitudinal and transversal) without taking into account inertia forces corresponding to the coordinates,
- variability of normal and tangential reactions between the wheels of the car and the surface (model of cooperation of the rubber wheel with the surface by Dugoff, Fancher, Segel [11, 12] later modified by Uffelman [13]),
- anti-lock braking system ABS.

The choice of such a model was determined by the range of interesting parameters of the research object and the scope of research. A model with a similar degree of complexity was used in [12], where the results of computer and research calculations were sufficiently compatible, even though it was also a quasi-spatial model.

The model adopted some simplifications, which assume, among others, that:

- the road surface is perfectly flat, horizontal and non-deformable,
- the impact of the drive system on the road wheels is taken into account in the form of driving moments,
- the characteristics of elastic elements of suspensions and tyres will be linear,
- pivot axle is perpendicular to the road surface,
- the offset radius will be changed by moving the wheel (changing the track width).

Figure 4 shows the scheme of the analysed system.

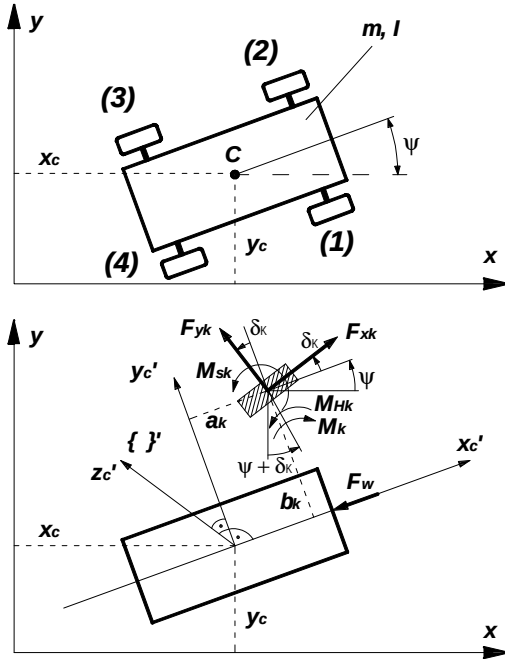


Fig. 4. Forces and moments acting on the vehicle [14]

$$m\ddot{x}_c = \sum_{k=1}^4 F_{xk} \cos(\psi + \delta_k) - \sum_{k=1}^4 F_{yk} \sin(\psi + \delta_k) - F_w \cos \psi \quad (1)$$

$$m\ddot{y}_c = \sum_{k=1}^4 F_{xk} \sin(\psi + \delta_k) + \sum_{k=1}^4 F_{yk} \cos(\psi + \delta_k) - F_w \sin \psi \quad (2)$$

$$I\ddot{\psi}_c = \sum_{k=1}^4 \left\{ -b_k [F_{xk} \cos \delta_k - F_{yk} \sin \delta_k] + a_k [F_{xk} \sin \delta_k + F_{yk} \cos \delta_k] + \sum_{k=1}^4 M_{sk} \right\} \quad (3)$$

$$I_k \ddot{\phi}_k = M_k - M_{Hk} - F_{xk} \cdot r_{dk} + F_{zk} \cdot f'_k \quad (k = 1, 2, 3, 4) \quad (4)$$

where: F_{xk} , F_{yk} , F_{zk} – component forces of the interaction between road surface and the wheel ($k = 1, 2, 3, 4$), M_{sk} – aligning moment, F_w – air drag, m – vehicle mass, I – the mass moment of vehicle inertia about the z -axis, ψ – yaw angle, δ_k – steering angle of the wheel (for rear wheels $\delta_3 = \delta_4 = 0$ is assumed), I_k – the mass moment of inertia of the wheel about its axis of rotation, ϕ_k – wheel rotation angle ($k = 1, 2, 3, 4$), M_k – drive moment, r_{dk} – dynamic radius of the wheel, f'_k – substitutive coefficient:

$$f'_k = \frac{F_{xk} - f_k \cdot F_{zk}}{C_{op}}$$

M_{Hk} – braking moment, f_k – rolling drag coefficient, C_{op} – tyre peripheral stiffness index (the same for all wheels). In addition, it was assumed that for $k = 1, 2, 3, 4$ forces acting on wheels in the plane of the roadway, and aligning moments, are expressed by the following formulas:

$$F_{xk} = \alpha_k \cdot N_k \quad (5)$$

$$F_{yk} = \beta_k \cdot N_k \quad (6)$$

$$M_{sk} = \gamma_{xk} \cdot F_{xk} + \gamma_{yk} \cdot F_{yk} + \gamma_{ko} (F_{xk} F_{yk}) = \gamma_{ko} + \gamma_k \cdot N_k \quad (7)$$

where: $\gamma_k = \alpha_k \cdot \gamma_{xk} + \beta_k \cdot \gamma_{yk}$, $\gamma_{ko} = \gamma_{ko} (F_{xk} F_{yk})$, has a value calculated in the previous step during the integration of the equations of motion, $N_k = F_{zk}$ – normal forces acting on the wheel $k = 1, 2, 3, 4$, α_k , β_k , γ_{xk} , γ_{yk} , γ_{ko} – coefficients calculated for the adopted model of cooperation of the wheel tyre with the road surface according to Dugoff [11].

Based on the adopted vehicle model, a computer program was developed. It allows you to define any road surface configurations under the wheels of the tested vehicle. It also allows you to simulate the operation of the ABS system. For this purpose, an appropriate algorithm that allows the implementation of the functions of this system, has been developed. There is also the possibility of requesting the desired drive and braking torque for individual wheels, as well as the actual driving and braking torque during road tests.

5. Simulation tests

5.1. Speeding the vehicle straight ahead on a road with homogeneous high adhesion coefficient

During the simulation test in speeding the vehicle, the steering wheel was maintained with a zero steering angle. The same driving torque was assumed for a vehicle equipped with an internal combustion engine and a vehicle with electric motors in wheel hubs, with the difference that in the case of vehicles with four-wheel drive, the torque was divided 50:50 for axles: front and rear.

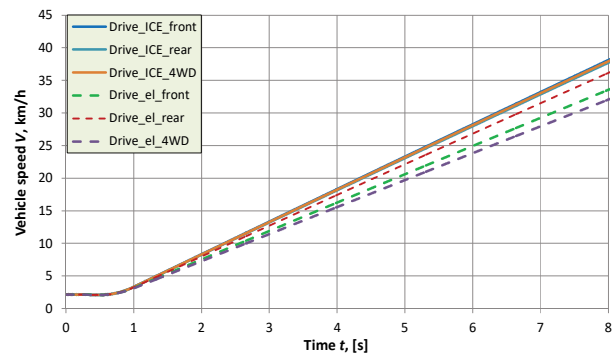


Fig. 5. Vehicle speed in speeding the vehicle straight ahead on a road with homogeneous high adhesion coefficient (ICE – internal combustion engine, el – electric motor) [own research]

Figure 5 shows the speed of a vehicle equipped with various propulsion systems configurations. The graphs show that the vehicle with classic drive achieves higher end speeds (after 8 second). The vehicle with electric motors in the wheel hubs, due to the higher weights, achieves lower end speeds, especially for the 4x4 drive system. A car with elec-

tric motors in the rear wheel hubs accelerates to a higher speed than with the electric motors in the front wheel hubs.

To assess the acceleration of the vehicle, the coefficient of acceleration intensity by analogy to the braking ratio coefficient of the vehicle was used, which is the ratio of vehicle acceleration to gravity acceleration.

Figure 6 presents the average values of the acceleration intensity coefficient of the vehicle equipped with various configurations of propulsion systems. The graphs show that a vehicle powered by an internal combustion engine achieves similar acceleration intensities (~ 0.14) regardless of whether it is an front or rear drive system, or all wheels drive system. In the case of vehicles with electric motors in the wheel hubs, the vehicle with electric motors mounted on the rear axle gets the highest acceleration intensity, the smaller ones are obtained in the case of electric motors on the front axle and electric motors driving all the wheels.

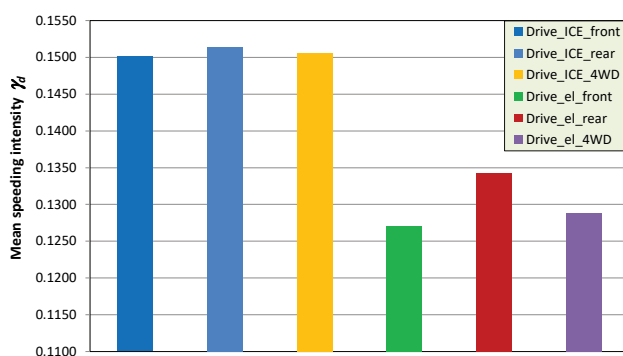


Fig. 6. Mean speeding intensity in speeding the vehicle straight ahead on a road with homogeneous high adhesion coefficient [own research]

5.2. Acceleration of the vehicle while driving straight ahead on the road surface with different coefficients of adhesion under the wheels of the right and left side of the vehicle

Similarly to the simulation tests, speeding at the road with uniform coefficient of adhesion, on the road with different coefficients of adhesion on the left and right side of vehicle, the steering wheel was maintained with a zero steering angle. Similarly, the same driving torque was assumed for a vehicle powered by an internal combustion engine and electric motors, with the difference that in the case of vehicles with all-wheel drive, the drive torque was divided in proportion 50:50 on each axle.

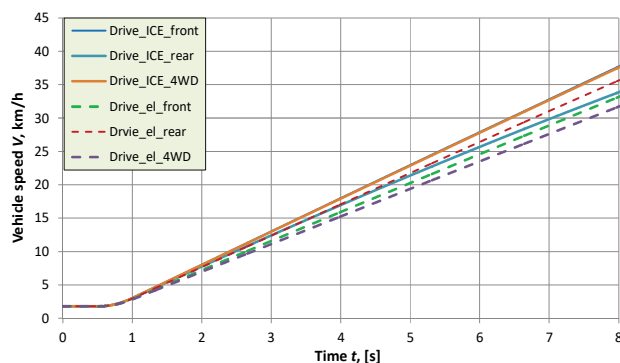


Fig. 7. Vehicle speed in speeding the vehicle straight ahead on a road with different adhesion coefficient under the wheels of the left and right side [own research]

Figure 7 shows the speed of a vehicle equipped with various configurations of propulsion systems, comparing its motion on the road with different coefficients of adhesion under the wheels of the right and left side of the vehicle. The graphs show that the vehicle driven by the internal combustion engine achieves slightly higher final speeds (after 8 second), with the vehicle with the rear drive showing a fairly large deviation from the rectilinear direction of travel. The vehicle powered by electric motors in the rear wheel hubs is characterized by a similar, though slightly lower, speed after 8 seconds of acceleration. Vehicles with engines in wheel hubs, due to the higher moments of inertia of the drive wheels, obtain lower final speeds, especially with the 4x4 drive system. A car with electric motors in the rear wheel hubs accelerates to a higher speed than with electric motors in the front wheel hubs.

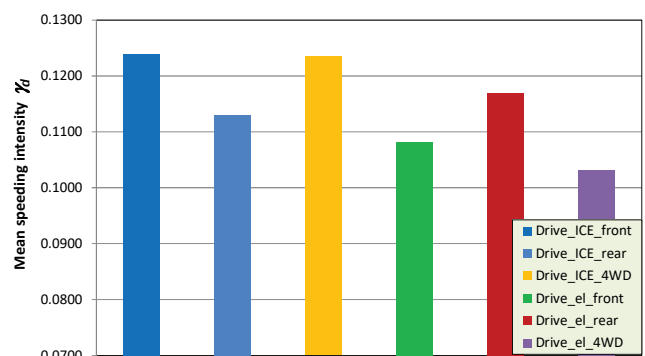


Fig. 8. Mean speeding intensity in speeding the vehicle straight ahead on a road with different adhesion coefficient under the wheels of the left and right side [own research]

Figure 8 presents the average values of the acceleration intensity of the vehicle equipped with various configurations of propulsion systems. The charts show that the vehicle powered by the internal combustion engine achieves similar acceleration intensities (~ 0.12) for the front drive system and the all-wheel drive. Lower values are obtained with a rear-wheel drive vehicle.

In the case of vehicles with electric motors in the wheel hubs, the vehicle with engines on the rear axle is characterized by the highest acceleration intensity, the smaller ones are obtained in the case of electric motors mounted on the front axle and electric motors driving all wheels.

5.3. Braking the vehicle while driving straight ahead on a road surface with a high homogeneous coefficient of adhesion

During braking simulation tests, the steering wheel was maintained with a zero steering angle. The same braking torque was assumed for a vehicle equipped with an internal combustion engine and a vehicle with electric motors in the wheel hubs, with the difference that in the case of vehicles with all-wheel drive, the braking torque was divided in proportion 50:50 for any axle. The anti-lock braking system (ABS) was used during braking.

Figure 9 shows the changes in vehicle speed as a function of the distance traveled to its stop. The vehicle was equipped with various configurations of drive systems. The graphs show that the road to stop the vehicle powered by

the internal combustion engine is slightly shorter than that for a vehicle equipped with electric motors in the rear axle wheel hubs. This one is shorter than the road to stop in the vehicle with electric drive motors on the front axle. A vehicle with electric engines mounted in all wheels was characterized that it has the longest way to stop. The moments of inertia of the wheels affect the length of the braking distance when the ABS system was used.

Figure 10 presents the average values of the braking ratio of a vehicle equipped with various configurations of propulsion systems. The graph shows that the vehicle powered by the internal combustion engine is characterized by the highest braking ratio (~ 0.670) for the system with the front drive and electric motors in the wheel hubs (~ 0.664), for the rear drive system and electric motors in the wheel hubs (~ 0.665) and with electric motors in the hubs of all wheels (~ 0.652).

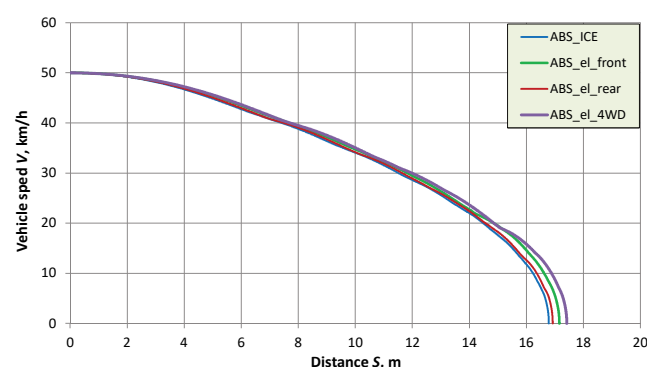


Fig. 9. Distance of stopping the vehicle in braking the vehicle straight ahead on a road with homogeneous high adhesion coefficient [own research]

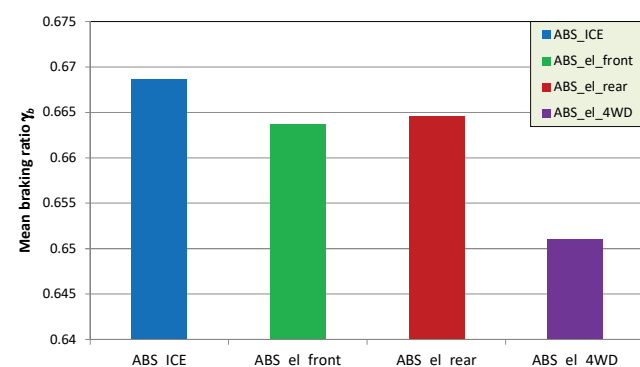


Fig. 10. Mean braking ratio in braking the vehicle straight ahead on a road with homogeneous high adhesion coefficient [own research]

5.4. Braking of the vehicle while driving straight ahead on the road surface with different coefficients of adhesion under the wheels of the right and left side of the vehicle

Similarly to the other simulation tests, the steering wheel was maintained with a zero steering angle. In these tests, the anti-lock braking system (ABS) was also simulated.

Figure 11 shows changes in vehicle speed as a function of the distance traveled to its stop. The graph shows that the road to stop the vehicle with the combustion engine used to drive and for the vehicle equipped with electric motors to drive mounted on the front axle is almost the same. The

vehicles with electric motors in the rear wheel hubs and in the hubs of all wheels were characterized by a little more stopping distance.

Figure 12 presents the average values of the braking ratio of a vehicle equipped with various configurations of propulsion systems. The graphs show that the vehicle powered by electric motors in the front wheel hubs has the highest braking ratio (~ 0.397) for a vehicle powered by an internal combustion engine (~ 0.387), for a vehicle powered by electric motors placed in the rear wheel hubs and with electric motors in the hubs of all wheels is (~ 0.337).

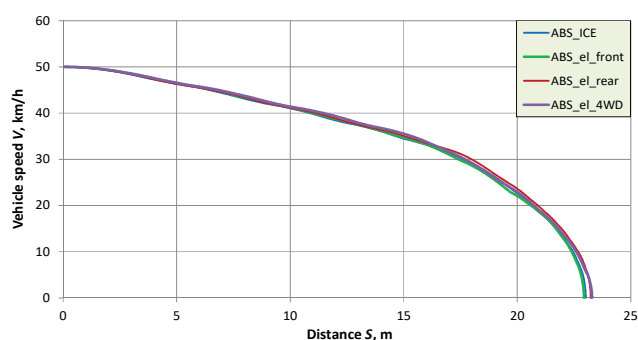


Fig. 11. Distance of stopping the vehicle in braking the vehicle straight ahead on a road with different adhesion coefficient under the wheels of the left and right side [own research]

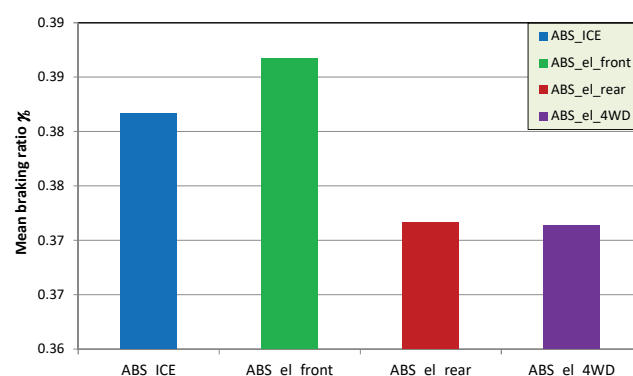


Fig. 12. Mean braking ratio in braking the vehicle straight ahead on a road with different adhesion coefficient under the wheels of the left and right side [own research]

6. Analysis of the behavior of the vehicle on the road with the same and different coefficient of adhesion under the wheels of the right and left side of the vehicle

Comparing of accelerating of the vehicle on a straight path of road with a high coefficient of adhesion ($\mu = 0.8$) with acceleration on the road with different coefficients of adhesion under the right wheels ($\mu = 0.3$) and the left side of the vehicle ($\mu = 0.8$) it was noticed that in the first case the distance traveled in the first 8 seconds was in the range from 37 to 43 m and in the second case from 32 to 37 m. Larger moment of inertia of the wheels with electric motors mounted in hubs affect the speed after 8 second of vehicle movement. The mean acceleration intensity of the vehicle driven by combustion engine, without the increased weight of the wheels and without the larger moments of inertia is almost constant and amounts to ~ 0.14 . After the installation

of electric motors in the wheels, mean acceleration intensity is lower, from ~ 0.12 to ~ 0.135 .

The acceleration on the road with different coefficients of adhesion results in a reduction of the mean acceleration intensity from ~ 0.105 to ~ 0.112 . The smallest acceleration intensity is characterized by the solution of the drive system with the motors located in the hubs of all wheels.

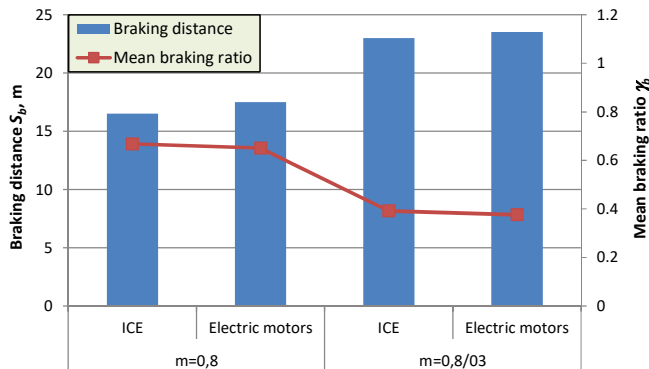


Fig. 13. Comparison of braking in an attempt to drive straight ahead on a road with high homogeneous and different adhesion coefficient under the wheels of the left and right side [own research]

The braking distance to stop of the vehicle on a straight path of road with a high coefficient of adhesion is from 16.5 m to 17.5 m. In the case of braking on the road with different coefficients of adhesion, it is extended to 23–23.5 m. The braking ratio in first case is from ~ 0.668 to ~ 0.651 . Braking on the road with different coefficients of adhesion reduces the braking ratio to 0.392–0.367.

The vehicle with electric motors placed in the hubs of all wheels is characterized by the lowest braking intensity indices. It is caused by the increase of vehicle mass and wheels mass as well as moments of inertia of the wheels, which change their rotational speed during the operation of the ABS system.

Figure 13 presents a comparison of vehicle braking with and without electric motors in wheel hubs on road surfaces with a high coefficient of adhesion and different coefficients of adhesion under the right and left wheels of the vehicle. Mounting electric drive motors in the vehicle's wheel hubs has little effect on increasing the braking distance and changing the braking ratio. A much greater im-

pact is observed in braking on the case of road with different coefficients of adhesion (μ -split).

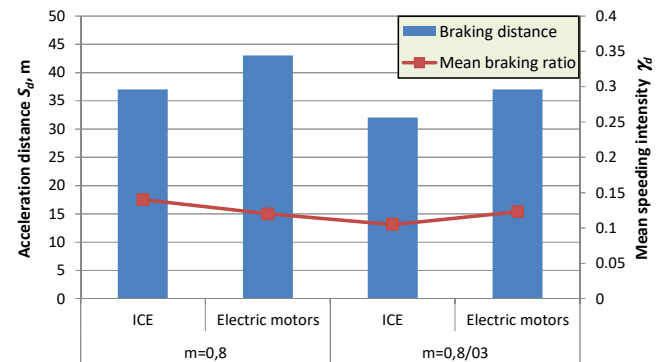


Fig. 14. Comparison speeding the vehicle in an attempt to drive straight ahead on a road with high homogeneous and different adhesion coefficient under the wheels of the left and right side [own research]

Figure 14 presents a comparison of acceleration of vehicles equipped without and with electric motors in wheel hubs on the road surfaces with a high coefficient of adhesion and different coefficients of adhesion under the wheels of the right and left side of the vehicle. Mounting drive motors in the wheel hubs of the vehicle contributes to the extension of the acceleration path and the reduction of its intensity. The influence of different coefficients of adhesion (μ -split) on the length of the acceleration path and its intensity is less than in the case of braking the vehicle.

7. Summary and conclusions

The paper presents preliminary simulation tests of a vehicle powered by electric motors placed in wheel hubs. For comparison, a vehicle with an internal combustion engine (smaller wheel masses and their moments of inertia) was also analyzed. The analysis was based on calculations using the original program described in the literature [14].

The use of electric motors placed in the wheel hubs for driving the vehicle reduces the driving comfort, but also changes the moment of inertia of the wheels. Especially during acceleration and intensive braking, the increase of masses and moments of inertia of the wheels will result in lowering both the intensity of the acceleration and the braking intensity of the vehicle.

Nomenclature

ABS anti-lock braking system

μ -split different coefficients of adhesion under the right side ($\mu = 0.3$) and the left side of the vehicle ($\mu = 0.8$)

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Tests of the vehicle's power unit with fuel cells at a reduced ambient temperature

The article presents the results of preliminary tests of the vehicle's powertrain with fuel cells, carried out in a thermoclimatic chamber in a reduced temperature box. The aim of this research is to facilitate the introduction of this type of vehicle to the market and guarantee correct operation in changing climatic conditions. In tests, the use of a new type of vehicle drive system and the use of hydrogen as a commonly available fuel is of great importance.

Key words: fuel cell, hydrogen, thermoclimatic chamber, low temperature

1. Introduction

The search for new sources of propulsion, which in the near future may replace piston combustion engines is currently a priority in the work of research teams dealing with automotive vehicles. New propulsion sources must meet a number of requirements related to the expectations of customers accustomed to classic propulsion systems, in addition, they must be simple and cheap to operate, and must not pollute the environment. Vehicles with electric drive, in which electric energy is stored in batteries, do not meet the aforementioned conditions, although they are currently very preferred. At the current stage of development they can only serve as urban vehicles, but they are not suitable for intercity traffic as well as for the drive of heavy commercial vehicles or construction machines. The transitional phase is vehicles with hybrid propulsion systems that only partially meet today's requirements. In this situation, only the propulsion system, equipped with hydrogen-powered fuel cells, can compete with classic reciprocating diesel engines. Currently, only few manufacturers offer vehicles with this type of propulsion system. Their widespread introduction to the market requires proper preparation, the important step of which will be comprehensive research conducted in various operating conditions. One of the most difficult conditions for using a fuel cell drive unit is the winter period, when the ambient temperature decreases below 0°C. This is due to the fact that during operation the water is released as a result of combining hydrogen with oxygen, therefore the operation of hydrogen fueled fuel cells at low ambient temperatures requires special procedures.

European Union countries belong to the group of countries where vehicles equipped with a new type of propulsion source will first appear on the market. Therefore, preparatory measures should be taken now to facilitate their market introduction and guaranteeing proper operation. What matters here is not only the use of a new type of propulsion system, but also the use of hydrogen as a commonly available fuel. An important factor that should be considered here are climate conditions. In a large part of Europe there is a temperate climate, which is characterized by high variability of temperature and humidity in particular seasons. In the case of operation in this area of vehicles equipped with hydrogen fuel cell drive, it is very important that the vehicle behaves in winter, when the temperature reaches a mini-

mum value of around -20°C, and there are large temperature fluctuations in the 24-hour cycle. In low ambient temperature, the working conditions of many structural elements of the vehicle's propulsion system are changed, related to the phenomenon of thermal expansion, and also to the phenomenon of condensation and changes in density of gaseous factors, as well as freezing of liquid agents. The mentioned phenomena are particularly important in the propulsion system of a vehicle equipped with fuel cells, where the fuel used is hydrogen, which requires appropriate conditions to maintain the tightness of the power supply system. In addition, the water drainage system resulting from the reaction of hydrogen with oxygen in the fuel cell requires special thermal conditions and purification procedures.

In connection with the above, before the introduction of this type of vehicles to the market, relevant from the point of view of operational safety, will be research conducted in various climatic conditions, especially at low ambient temperature.

2. The scope of tests of a vehicle powered by fuel cells

At the Cracow University of Technology a preliminary vehicle was equipped with hydrogen-powered fuel cell drive. The object of the research was a Toyota Mirai car in a standard version, equipped with an electric drive system and fuel cells, supplied with hydrogen with a purity of 99.97% H₂ stored under 700 bar pressure in two tanks made of composite materials. These tests were carried out in an accredited thermoclimatic chamber that allows a quick change in the ambient temperature. The chamber with dimensions: 23.7×6.9×6.5 m, belonging to the Thermoclimatic Research Laboratory of the Cracow University of Technology, allows testing of various types of facilities, including vehicles, in the temperature range from -50 to +70°C.

The main goal of the research was to gain operational experience, allowing for better market preparation for the widespread introduction of this type of vehicles into operation. The scope of the research has been limited to a preliminary analysis of the operation of individual construction units of the power unit at a reduced temperature, in particular the hydrogen supply system for fuel cells. An important goal here was to pay attention to the tightness of the entire hydrogen supply system to the fuel cell during the change

of operating temperature. The second important goal was to assess the thermal insulation of the key components of the propulsion system, including in particular the fuel cell, hydrogen tanks and steam evacuation system, carried out by analyzing the change in temperature.



Fig. 1. Toyota Mirai during tests in the thermoclimatic chamber

3. Procedures used when operating fuel cells at low ambient temperature

During the operation of a fuel cell vehicle, manufacturers foresee the use of special procedures to protect the fuel cell, the power supply system and the antifreeze outlet system. When operating a Toyota Mirai at an ambient temperature below -10°C , certain activities are provided to protect the cells and other systems against damage. The diagram of these activities in various operating states of the drive system is shown in Fig. 2.

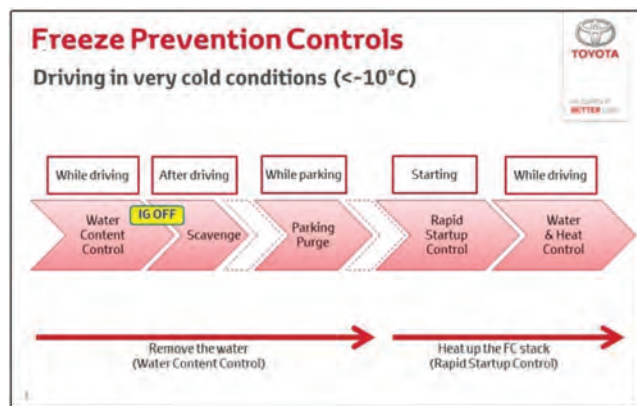


Fig. 2. Diagram of procedures for protecting the fuel cell of the Toyota Mirai at reduced ambient temperature [6]

The most important purpose of these special procedures is to remove water and dry the fuel cell components. No matter what the ambient temperature is, the fuel cell stack temperature is monitored every hour. If the fuel stack temperature gets below 0°C , a parking purge is performed and the air compressor, hydrogen pump and discharge valve are activated for up to 3 minutes in order to purge the water from mainly the anode side. In this case water builds up in the anode after switching off ignition due to condensation. When operating a fuel cell at a reduced ambient temperature, the excess water accumulating in the fuel cell and outlet system must be constantly removed. If the ambient temperature is below -10°C special cold spec purge will take place. This special cold spec purge is different than the

normal purge in the way that the water purge is controlled in a much very precise way.

The cold purge will take around 40 seconds and will dry the cathode effectively. But, even with a dry stack, at -30°C , it is possible that the very first water drops will immediately freeze. Therefore we need the so-called Rapid Startup Control that makes sure that this does not happen.

When the electronic control unit of fuel cell determines the system temperature is 0°C or lower or ambient air temperature is -10°C or lower while the vehicle is left alone, the air amount supplied to the fuel cells is restricted and the a large amount of heat is generated by the fuel cell stack assembly to warm up the system and prevent water from freezing. During the quick warm up, the fuel cell current amount is larger than normal even at the same output as power generation is prioritized during driving. As a result, the fuel cell air compressor and hydrogen pump operate at a higher speed than normal causing a noise to be produced.

In relation to the operation of standard piston combustion engines, the described activities related to fuel cell operation are completely different. Therefore, research to prepare users for the operation of new powertrains is warranted.

4. Research methodology

Research on the Toyota Mirai was conducted in a thermoclimatic chamber in the temperature range from -10°C to -18°C , which corresponds to the average temperature occurring in winter in the temperate climate zone. The vehicle was inserted into the chamber with an unheated link, and the test time was about 2.5 hours.

Measurements of many different physical quantities characterizing the operation of the drive unit were carried out, using signals available in the vehicle's on-board diagnostic system, read from the factory diagnostic system of Toyota. Particular attention has been paid to those parameters that have an impact on the operational safety of the drive system at a reduced temperature. The following parameters were measured:

- Measurement time,
- Temperature in the thermoclimatic chamber,
- Hydrogen pump motor temperature,
- Smoothed value of hydrogen tank 1 temperature,
- Smoothed value of hydrogen tank 2 temperature,
- FC stack coolant temperature (FC Stack Outlet),
- Smoothed value of fc stack coolant temperature (Radiator Outlet),
- FC stack coolant temperature (Radiator Outlet),
- Smoothed value of intake air temperature,
- Smoothed value of fc stack air temperature (FC Stack Inlet),
- Hydrogen pressure in high, medium and low pressure circuits.

5. Investigation results

During the measurements lasting about 2.5 hours, every hour the procedure of checking the condition of fuel cells was switched on, and in addition it was necessary to perform maintenance activities related to the need to enter service inside the chamber. This is manifested by temporary

changes in the temperature in the chamber. Therefore, the graph additionally includes a trend line, which indicates a continuous decrease in the temperature in the chamber, which during the tests varied from approx. -11°C to approx. -18°C .

The most sensitive element of the propulsion system is the Toyota Mirai car for lower temperatures are the individual segments of the fuel cell stack. During the tests in the thermoclimatic chamber conducted in the temperature range from -11°C to -18°C in about 2.5 hours, the temperature of the fuel cell liquid cooler cooled very slowly and during the tests was above 0°C , and its sudden increase took place with the activation of the cells (Fig. 3). A similar change was shown in the air temperature surrounding the stack of cells, where its value was significantly lower, which resulted from the existing air exchange between the interior of the vehicle and the thermoclimatic chamber (Fig. 3).



Fig. 3. The temperature of the cooling medium of the fuel cell stack and the air in the cell segments, during the tests of the vehicle in the thermoclimatic chamber

An important parameter describing the working conditions of the fuel cell is the temperature course in the fuel cell intake system (Fig. 4). The changes in the temperature values recorded on the chart are related to the flow of air



Fig. 4. Air temperature in the fuel cell intake system during vehicle tests in the thermoclimatic chamber

supplied to the fuel cell, both during its operation and during the rinsing procedure. The size of this change depends on the value of the air flow which travels from the thermoclimatic chamber to the inside of the cell.

When the vehicle is stationary at a temperature below 0°C , the procedure for cleaning the surface of the cells from the water is activated. This is manifested by the periodic switching on of the air compressor and the hydrogen pump in order to dry the surface. In Fig. 5. The temperature change of the engine driving the hydrogen pump is presented along with the temperature decrease in the thermoclimatic chamber. Visible, abrupt temperature changes are associated with the pump switching on, because its operation causes the gases to move and the cooling of its components. The sudden increase in temperature of the hydrogen pump seen in Fig. 4 occurred after switching on the fuel cell.

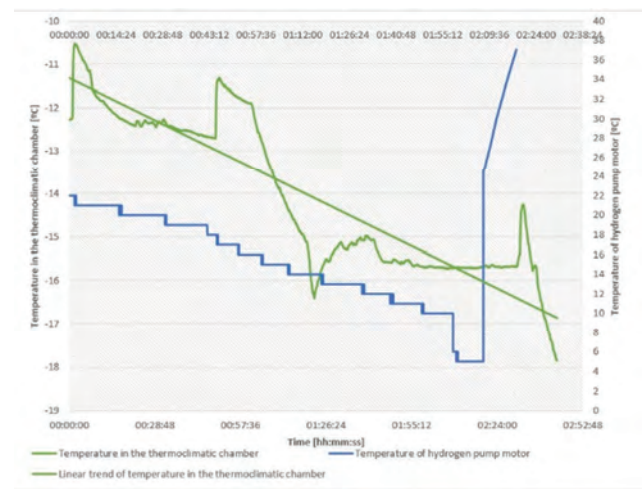


Fig. 5. Temperature of the hydrogen pump during testing of the vehicle in the thermoclimatic chamber

Figure 6 shows the temperature change of hydrogen tanks along with the temperature change in the thermoclimatic chamber. The sudden increase in temperature in the final part of the chart was related to the activation of fuel cells. The temperature of the hydrogen tanks changes much



Fig. 6. Temperature of hydrogen tanks during vehicle tests in a thermoclimatic chamber

more slowly than the temperature of the air in the thermoclimatic chamber, while a significant reduction in their temperature occurs with the start of the fuel cell and the outflow of hydrogen. This is *inter alia* related to the air movement around the hydrogen tanks.

During measurements in the thermoclimatic chamber, the pressure was also measured in individual circuits of the power system (Fig. 7). The high pressure circuit has included part of the power supply directly after the hydrogen tanks, the medium pressure circuit is responsible for supplying hydrogen to the fuel cells, while the low pressure circuit is already in the fuel cell itself. Despite the change in ambient temperature, the pressure in particular circuits had a stable value. The only significant change in pressure occurs when the cell is started when the hydrogen flowing out of the tanks undergoes heating processes.



Fig. 7. Hydrogen pressure in high, medium and low pressure circuits during vehicle tests in a thermoclimatic chamber

After the tests, all external parts of the vehicle were covered with ice, while the drive unit and installations were covered with moisture (Fig.8). Despite this, there were no problems with the starting of the vehicle's propulsion system and the operation of all components.

During and after the tests, attention was paid to phenomena that may be significant from the point of view of the operation of the vehicle with the drive system equipped with fuel cells. After completing the tests in the thermoclimatic chamber, all internal components of the vehicle were covered with water molecules. Due to the protection procedures of the fuel cell against the effects of low temperature, many electric circuits of the drive system maintain voltage, and the phenomenon of water condensation on these elements can lead to uncontrolled current flows. In this connection, attention was paid to the phenomenon of condensation of water vapor on individual vehicle systems, and especially on all types of electrical connections and the system of hydrogen supply from pressure cylinders to the fuel cell. In addition, a reduced temperature can affect the tightness of the hydrogen fuel supply system. There were no cases of freezing of moisture particles covering the outlet system, which may lead to the loss of the system's patency.

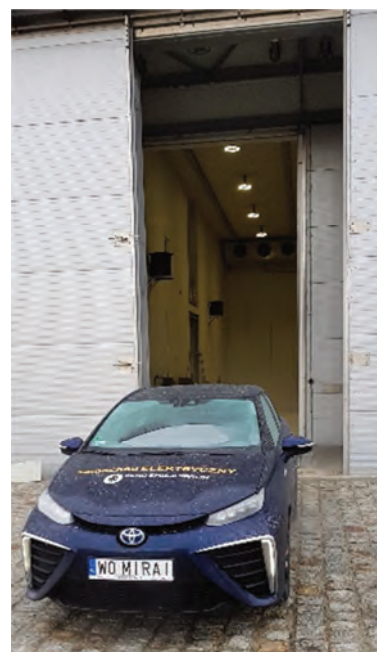


Fig. 8. Toyota Mirai after testing in a thermoclimatic chamber

A very important effect of the research in the thermoclimatic chamber was the opportunity to learn the course of procedures used in vehicles with FC drive system at a reduced ambient temperature.

6. Conclusions

- The operating principle and operating procedures of the vehicle's power unit with fuel cells are completely different to those known from power units with combustion engines,
- Preliminary operational tests are very important to help implement a new type of drive system for common use,
- Climatic conditions are a very important factor affecting the operation of fuel cells, therefore the tests carried out in the thermoclimatic chamber are fully justified and necessary,
- Because the factor released from the fuel cell into the atmosphere is water, it is particularly important to collect experience in the operation of this type of vehicle source at a reduced temperature,
- The tests have initial character and are used to identify phenomena and problems occurring during the operation of the power unit equipped with a fuel cell fueled with hydrogen,
- The operation of the vehicle's power unit, equipped with hydrogen fuel cells, in conditions of low temperature requires special procedures that are not used in other vehicle driving units,
- Particular attention has been paid to the operation of elements of the hydrogen supply system in conditions of low temperature,
- During the tests, it was found that many components of the drive system, including the live electrical connections and systems, were covered with a layer of condensed water during the tests. At the same time, no disturbances in their operation were found.

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The potential of thermoelectric energy harvesting in vehicles equipped with ICE

The paper deals with an issue of waste heat recovery in a selected configuration of an internal combustion engine. A possibility of using thermoelectric cells (currently available on the market) for production of electricity with heat extracted from the exhaust gas was considered. The calculations were made using specialized software. Features and design assumptions of the heat recovery system were presented and their influence on parameters of the entire system was investigated (efficiency of the internal combustion engine, power, etc.). An assessment of the applicability of the energy recovery system based on thermoelectric effects and characteristic of the proposed configuration was performed. Some issues that require further research have been highlighted.

Key words: combustion engine, heat recovery, thermoelectric module, heat transfer, exhaust system

1. Introduction

Improvement in internal combustion engine performance is one of the main goals engineers deal with. There are many sophisticated solutions based, in some sense, on controlling the combustion process inside the combustion chamber and the piston motion. Some of them increase pressure, temperature, compression ratio, valve timing, etc. But in nearly every combustion engine system a significant part of the energy is lost due to the waste heat expelled to the surroundings [10]. Some heat is lost in the engine cooling system, via the central radiator, but this heat source has relatively low temperature. Much higher temperature occurs, however in the exhaust system. An attempt was made to analyse the possibilities, issues and selected problems of heat recovery from the exhaust gas. One of the possibilities discussed in this paper is the use of the thermoelectric modules, which convert heat into electricity. The basic principle of their operation is the Seebeck effect. A thermoelectric module is usually a flat sheet (thickness of a few millimetres) of semiconductors between ceramic (electrical insulator) plates. If a thermal gradient occurs throughout the module a voltage difference is generated [3, 9]. The thermoelectric modules have several advantages like the lack of moving parts, long lifetime, very small size and noiseless operation. On the other hand, their efficiency is rather low. There are, of course, some more efficient methods of waste heat recovery (Organic Rankine Cycle, turbine) but they are usually quite complex and difficult or unreasonable to use in mobile applications [4, 14]. The thermoelectric module can be easily retrofitted, e.g. by replacing genuine muffler with a modified one. Descriptions of experimental research systems, including modules assembled into vehicles can be found in the literature [2, 7, 11]. The following considerations were performed to make a coarse assessment of the amount of heat which can be recovered, overall characteristics of such heat recovery system and different working conditions that can occur in typical vehicle drivetrain. The result obtained can be useful in a design process of specific heat exchangers, identifying crucial issues and targets.

2. Modelling approach

The analysis of waste heat recovery was based on a 4 stroke, a spark ignited, 500 ccm motorcycle engine. A sim-

plified model of immediate fuel combustion at a constant volume was used. Perfectly stoichiometric air to fuel ratio was assumed and friction inside the engine block between moving elements was neglected. The exhaust system in proposed configuration consists of, among the others, a thermoelectric module assembly unit. This assembly consist of a straight pipe, in which exhaust gases flow, a highly heat conductive material between inner pipe surface and the thermoelectric cells (hot side heat exchanger – heat source), the thermoelectric module and the cooling system. The hot side of the thermoelectric module can be attached directly to the heat source (a flat surface covered with thermal grease). The opposite side of the thermoelectric module should be attached to the water cooling system (cold side heat exchanger – heat sink). The remaining surfaces were assumed adiabatic. Figs 1 and 2 present both a cross-sectional and an isometric view of the heat recovery system proposed.

The specialized software – AVL Boost, was used to carry out the simulations.

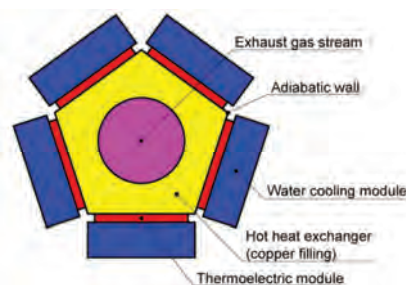


Fig. 1. A cross-sectional view of thermoelectric module assembly

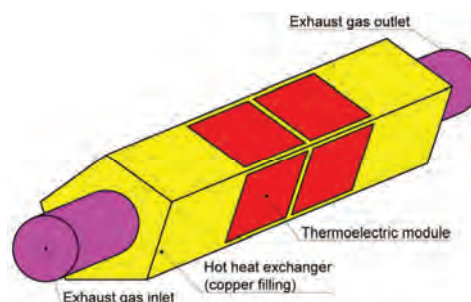


Fig. 2. Thermoelectric module assembly

The overall thermal resistance of this assembly can be calculated as the proportionality factor between heat transferred through the system (from the exhaust gas to cooling water) and temperature difference at both sides of the domain. One of them is the inner surface in contact with hot exhaust gas, whereas the second one is the thermoelectric module cold side (cooled with water flow from the cooling system). Due to a major difference in magnitudes of thermal resistance of the thermoelectric module and the heat exchangers the heat exchangers resistance can be neglected [1]. In other words, if constant heat flux is imposed to the system the temperature gradient is significant only across the thermoelectric module whereas within the hot heat exchanger the temperature distribution is nearly uniform. The same situation occurs at the cold side – the temperature of the heat exchanger and the module cold side is almost the same. This assumption is true for heat exchangers made of a highly heat-conductive material, e.g. copper. In such a case, the gas-wall temperature difference drop along the flow direction is not large. In the described configuration it is relatively small (not greater than 100 K), so the temperature of heat source for modules in one thermoelectric assembly was assumed to be constant. As a result, in one assembly the temperature of the inner wall (in contact with exhaust gases) and thermoelectric modules hot side is the same [1, 6].

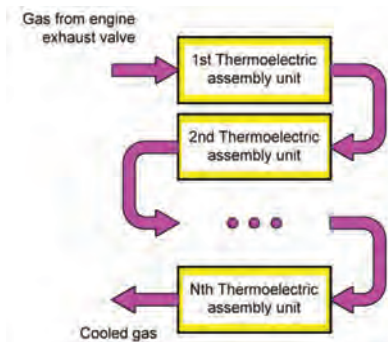


Fig. 3. General configuration of exhaust heat recovery system

Based on literature and the technical data of commercially available thermoelectric modules, the thermal conductivity of exemplary module (Tecteg TEG1-PB12690) was assumed as about 0.5 W/K ($\sim 273 \text{ W}/570^\circ\text{C}$) for surface $62 \text{ mm} \times 62 \text{ mm}$ [12] (0.5 W/K corresponds to 2 K/W thermal resistance). Thermal conductivity of the thermoelectric module is subjected to change with electrical load (electric current in a circuit results with the Peltier effect – phenomena related to the Seebeck effect). However, using a DC/DC converter within the system, whose aim is to provide optimal working conditions (electrical resistance of power receiver for the highest module power or the highest efficiency) immediately minimises the problem. So, the overall thermal conductivity can be assumed constant [1]. Proposed thermoelectric assembly unit consists of 10 mentioned thermoelectric modules attached to the exhaust pipe. The modules used have a maximum allowable working temperature of 600°C . To keep the constant temperature distribution within the heat source, the modules should be uniformly distributed on the heat source surface, e.g. if the

cross section is pentagon-like showed on Fig. 1 it results in 2 thermoelectric modules ($62 \text{ mm} \times 62 \text{ mm}$) on each wall. The temperature of the heat source was determined based on the 0.5 W/K thermal conductivity of the thermoelectric modules. For 10 thermoelectric modules (one assembly unit) the thermal conductivity (reverse of thermal resistance) is about 10 W/K. On this basis, a script in C language was developed and implemented in the software to calculate the temperature of the heat source. The module cold side temperature (resulting from water cooling) was assumed to be 50°C . To model the heat transfer coefficient between the stream of exhaust gas and the channel wall (heat source, with a temperature equal to module hot side) a Reynolds analogy have been used. It provides a rough approximation of a general heat exchanger. The amount of electric energy produced as useful result in thermoelectric module was calculated using the heat transferred from exhaust gas into the heat exchanger in assembly unit (heat recovered in a unit) multiplied by temperature gradient between hot and cold side of the thermoelectric module and then multiplied by factor (0.000128) corresponding to the datasheet of the thermoelectric module ($\sim 7\%$ efficiency at $600^\circ\text{C}/50^\circ\text{C}$). The efficiency of electric power generation can be assumed proportional to the temperature gradient, so it is adopted here.

An alternative method for efficiency assessment can be based on Carnot efficiency for given temperatures (heat source and heat sink) multiplied by an appropriate factor. This method corresponds to the assumption that the thermoelectric figure of merit (coefficient of performance) is constant over temperature for the thermoelectric module component materials (semiconductors, legs) [9].

3. Reference experimental setup

At the initial step of considerations, to avoid an influence of intensive wave phenomena, a large plenum (500 litres) have been added between the exhaust port and the thermoelectric module assembly(s). It was assumed adiabatic, so temperature and enthalpy changes were negligibly small. Its only task to stabilize the exhaust gas velocity to obtain reference data without the influence of pressure waves and cyclic motions. The plenum is connected with exhaust port via adiabatic, 31 mm – diameter and 500 mm – long pipe. The pressure waves within the pipe influence to some extent the exhaust gas discharge, but these phenomena are not analysed here. Dimensions of this pipe were chosen to reduce the unavoidable effects as much as possible. The variations in velocity due to the plenum over one engine cycle do not exceed 3%. Without the plenum, the changes are, not only bigger than 100%, but also a back-flow is observed. So, the use of plenum justifies the assumption of a constant velocity flow. A total number of nine thermoelectric assembly units were simulated. Figure 3 presents the overall view of the AVL Boost model. These units are represented by pipe 8 and 10–17 (marked in red in Fig. 3). Restrictors between them are dummy elements with coefficients “1”. The formula interpreters calculate wall temperatures for thermoelectric assembly unit (one pipe means one unit, each of the nine units is equivalent of 10 thermoelectric modules). All simulations were made to represent only the steady-state conditions and the calcula-

tions were conducted to achieve the convergence with the spatial resolution of 10 mm.

The heat flux throughout the assembly can be adjusted, within the software, with the heat transfer factor parameter. It is a ratio of real heat transfer coefficient and the value used in calculations. A value equal to one represents conduction in the simple pipe, as shown in Fig. 1. Values greater than 1 represent e.g. internally finned pipe or other changes in geometry which intensify the heat flow. This value was set to keep the maximum temperature at the allowable limit (exceeding 600°C of pipe wall temperature would damage the module). This condition results in heat transfer factor of 1.02. Particular solutions of heat exchangers for thermoelectric modules to provide the best heat exchange can be found in the literature [5, 8].

The calculations were made for a reference point - full throttle, 6000 rpm. The mechanical power of the engine is therefore 24.62 kW. Total enthalpy flow behind the plenum is 27.7 kW (reference zero: 1 atm; 25°C), which corresponds to temperature 1278 K, mass flow 24 g/s and velocity (mean) 69 m/s. Table 1 shows some selected results obtained for the thermoelectric module assembly units. Heat flux means heat recovered in particular assembly unit (heat flux through thermoelectric modules), electric power is generated by thermoelectric modules as described above, and it is the useful effect.

a reduction in the temperature of the exhaust gas which, in consequence, decreases the heat flux and efficiency of the thermoelectric modules due to a drop of temperature gradient.

Table 1. Selected parameters of nine units (heat transfer factor for every unit – 1.02) during operation in full throttle at 6000 rpm

Thermoelectric assembly unit	Inlet gas temp. [K]	Outlet gas temp. [K]	Wall temp. [K]	Heat flux [W]	Electric power [W]
1	1278	1191	873	2748	193
2	1191	1111	820	2483	158
3	1111	1038	771	2242	129
4	1038	972	728	2023	105
5	972	910	688	1823	85
6	910	855	651	1642	69
7	855	804	651	1478	56
8	804	758	589	1330	45
9	758	716	562	1196	37
Sum				16966	877

Table 2. Selected parameters of nine units for variable heat transfer factor during operation in full throttle at 6000 rpm

Thermoelectric assembly unit	Inlet gas temp. [K]	Outlet gas temp. [K]	Wall temp. [K]	Heat flux [W]	Electric power [W]	Heat transfer factor [-]
1	1278	1191	873	2751	194	1.02
2	1191	1102	873	2751	194	1.38
3	1102	1013	873	2751	194	2.10
4	1013	921	873	2751	194	4.49
5	921	836	827	2520	163	10
6	836	761	754	2154	119	10
7	761	697	754	1836	101	10
8	697	641	635	1561	62	10
9	641	593	588	1325	45	10
Sum				20400	1265	

In the next step, a PID regulator has been involved in the system and tuned to set the heat transfer factor for every thermoelectric module assembly independently. Obtained data are presented in Table 2.

As it can be noticed from Table 2, the total recovered energy in the form of electric power is about 1.27 kW. To provide the same heat flux, and in consequence, the same temperature of the module, greater heat transfer factors are required. This is because of the drop in enthalpy of the exhaust stream. Here, the heat transfer factor was arbitrarily limited to 10.

In the next stage, the plenum has been modified. Its volume changed from 500 l to 5 l. The variation of velocity during one engine cycle is now significant. Values of instantaneous velocity behind the plenum are in this case between 110 m/s forward and 10 m/s reverse. For this situation, similar analyses were made as described previously and presented in Table 2. The mean inlet and outlet gas temperatures are the same as before. For the first four thermoelectric module assemblies, the wall temperatures also must be the same because it is a criterion to tune the heat transfer coefficient and it is equal to the maximum allowable thermoelectric module temperature.

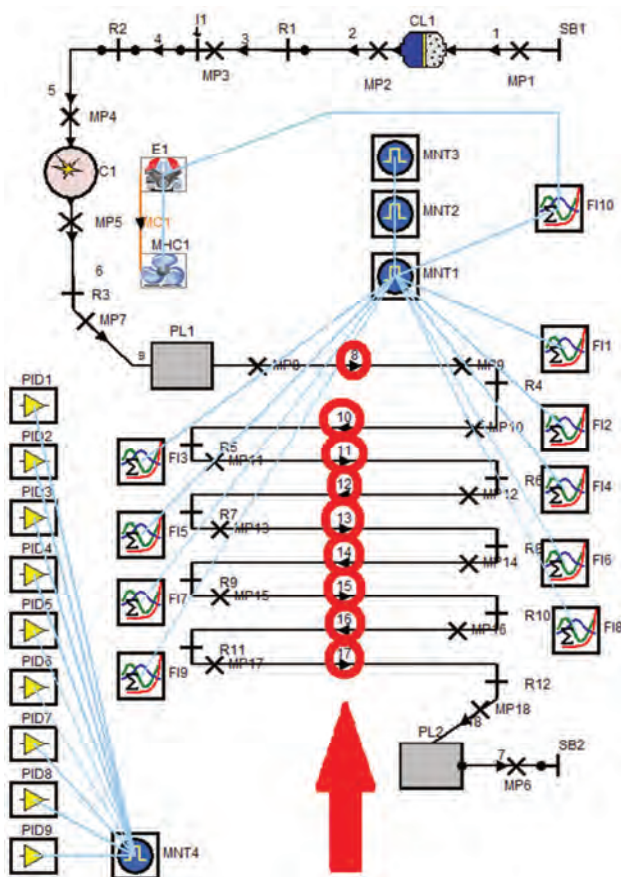


Fig. 4. Overall view of test bench in AVL Boost

It can be seen from Table 1, that efficiency of each next thermoelectric assembly unit decreases. This results from

Table 3. Optimal heat transfer coefficient for each first four subsequent thermoelectric module assembly

Thermoelectric assembly unit	1	2	3	4
Heat Transfer Coefficient	1.06	1.44	2.26	5.34

Because of the amount of energy and reasonably small heat transfer factor, further considerations are limited to four thermoelectric module assemblies. The heat transfer factors are also similar. Table 3 presents the heat transfer coefficients for the 5 l plenum.

4. Experimental setup in different working conditions

For further considerations, the system consisting of four thermoelectric module assemblies (each assembly consists of 10 modules) were taken into account. Each of them has different heat transfer factor coefficient presented in Table 3, which is optimum for the reference point (6000 rpm, full throttle). The proposed system was analysed at different working conditions.

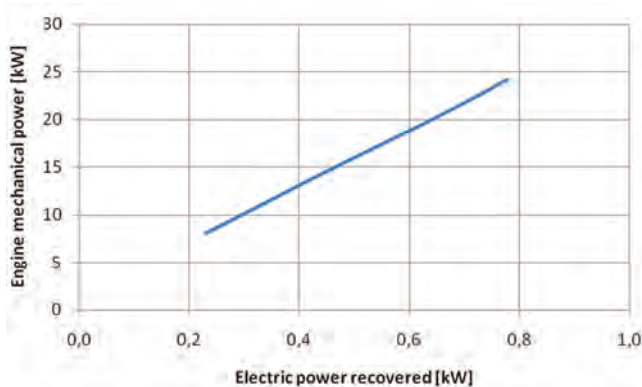


Fig. 5. Recovered power for 6000 rpm

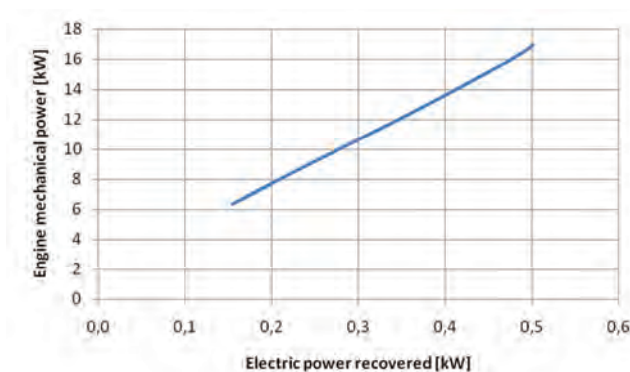


Fig. 6. Recovered power for 4000 rpm

Two rotational speeds of 6000 rpm and 4000 rpm were considered, with different throttle position, from 5% (flow coefficient restriction 0.05 in the intake manifold) to 100% (flow coeff. 0.98). Diagrams in Fig. 5 and Fig. 6 show the total amount of harvested electric energy in the four thermoelectric module assembly (40 thermoelectric modules in total).

5. Results discussion

The results of the proposed configuration have been presented above. In the first configuration, when nine thermoelectric module assemblies are identical, the recovered electric power was 877 W. It is about 3.5% of the engine mechanical output power. After modification of the thermoelectric module assembly assuming individual tuning of geometry (heat transfer factor) for every unit, the efficiency of more than 5% was achieved. But for units number 5 to 9 a high heat transfer factor is necessary. This value has been limited to 10 in this paper due to practical reasons. The highest possible amount of produced electric energy is achievable only in the first four units. The efficiency of the thermoelectric module becomes lower as the temperature of the heat source decreases. The influence of cyclic motion and pressure wave is relatively small in this case. It is hard to predict how this phenomenon influences the whole system. On one hand, the forward and reverse cyclic motion intensify the heat transfer factor, but on the other, some portion of the hot gases can be rapidly pushed through the heat exchanger. Some description and analysis of this phenomena can be found in [13]. The behaviour of the proposed system is nearly linear with decreasing load of the engine for both investigated rotational speeds.

6. Conclusion

Results obtained in the investigation can be useful for further analysis concerning the geometry of heat exchanger, economic profitability of waste heat recovery systems. It can also be helpful in rough estimation of the number of thermoelectric assemblies or simple modules. The geometry was assumed arbitrary. In the real arrangement, this can vary significantly and influence heat transfer phenomena due to cyclic pulsation in the exhaust system. It can enhance or lower the effective heat flux. The length of the heat exchanger can also have a different influence depending on the engine rotational speed and a ratio between gas pulse length and heat exchanger length. Some ways of enhancing the heat flux are the elongation of the heat exchanger, internal fins, turbulisers, etc. It has to be a subject of further investigation which one is more profitable. The acoustic phenomena have not been taken into account in this case and further investigation dealing with sound level and exhaust after treatment (3 W cat, DPF) should be also performed. The amount of energy possible to recover is not very large, Although it should be notified, that this is the electric energy – with high exergy level. In conventional solutions, the alternator is used to produce electricity and its efficiency, the amount of consumed mechanical power have to be also considered. In the proposed configuration, the power of alternator changes, it can possibly be switched off or working like an engine. The transient phenomena and heating period have to be also the subject of further investigation. Reasonability of using the thermoelectric solution strongly depends on the development of thermoelectric cells, materials and their cost.

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Experimental research of two stroke aircraft diesel engine

This paper presents the results of experimental studies of the opposed-piston diesel engine. This engine was designed during one of the stages of the research on a new-type drive unit for gyrocopter applications. In order to conduct research, a special test stand as well as control and measurement systems were developed. As part of the work on the engine, the fuel injection system, engine temperature control system and measurement systems were designed. In addition, a computer program has been developed for the fuel injection system control (injectors, valves fuel pressure regulators). The paper presents the results of the preliminary tests for a single value of engine speed (1500 rpm) and three values of load defined by torque. The measured value of the indicated pressure made it possible to calculate the maximum pressure. The results obtained from the bench tests were analyzed.

Key words: diesel, engine, indicated pressure, opposed-piston, two-stroke

1. Introduction

Spark ignition engines are widely used for propulsion of light aircrafts. In recent years, several diesel models (Thielert, SMA Engines) have appeared on the aviation market, which effectively compete in this field with SI engines. The innovative solution is the use of two-stroke opposed-piston diesel engine for propulsion of aircraft. A completely new type of propulsion requires a series of tests of such an engine.

The basic method of testing internal combustion engines are workplace tests using an electric brake and a set of measuring sensors. One of the key measurement signals in the engine test stands is the pressure signal in the cylinder. It can be used to create an open indicator graph. A comparison of a number of such ranges for subsequent work cycles allows for the creation of a dense indicator diagram. Its analysis makes it possible to assess the phenomenon of the uniqueness of subsequent work cycles, which may result from unequal filling of the cylinder with a fresh charge or an uneven combustion process. In addition, the pressure signal in the cylinder allows for a series of additional analyzes: combustion process, heat load and heat efficiency, mechanical load. On the basis of the pressure data for the gas in cylinder it is also possible to calculate the work performed by the gas acting on the piston [1]. Many publications focus on the analysis of the combustion process using cylinder pressure. a method for performing a quick thermodynamic diagnosis of the combustion process in a DI diesel engine. Pressure signal analysis in time and time-frequency domains can be used for detecting the start and the end of combustion and the heat release peak [3]. Moreover, the misfire phenomenon can be detected using the pressure cylinder signal [4]. The analysis of cylinder pressure allows for the investigation of start of combustion and calculation of an accurate net rate of heat release indicator diagram in the case of the dual-fuel engines study [5]. The cylinder pressure signal can also be used to analyze the noise of working engine. An example is work [6], in which the method of analyzing noise generated during the combustion process in a diesel engine was presented. A similar issue was analyzed in the paper [7]. The pressure cylinder can be used to control the operation of the engine. This method

was analyzed in the paper [8], which presents the control of the closed-loop combustion process using the digital cylinder pressure signal in a diesel engine. A similar solution was analyzed in the paper [9], where the cylinder pressure-based control system (CPBC) for conventional diesel combustion with high EGR levels was used. The authors of the paper [10] presented the use of cylinder pressure to control pre-mixed diesel combustion. The authors paid attention to the control effectiveness of the real-time cylinder pressure feedback.

In order to obtain an indicator diagram it is necessary to use the following devices: a pressure sensor placed in the cylinder, a position sensor of the crankshaft and a recorder. The direct application of the pressure transducer in the cylinder is limited for technical and economic reasons. Direct measurement requires a high-performance pressure transmitter capable of withstanding the difficult operating conditions inside the cylinder. Flame front transducers are expensive, have a limited lifetime and require some modifications to the design of the engine for their installation [11]. The authors of the publication [11] introduced the non-parametric modelling techniques to reconstruct the cylinder pressure in the diesel engine. In paper [12], an attempt was made to reproduce the cylinder pressure signal based on externally measured engine vibration signal.

The cylinder pressure signal can also be used to analyze exhaust emissions. In [13], a neural network was used to predict emissions of the direct injection diesel engine. The authors of the paper [14] presented a method of real-time estimation of engine-out particulate matter emissions of a diesel engine using in-cylinder pressure signal.

The purpose of the research was to conduct a preliminary analysis of the unevenness of work of the newly-designed two-stroke opposed-piston diesel engine. For this purpose, experimental tests were performed on a test bench. For the defined operating points, the value of pressure as a function of time was obtained in one of the cylinders of the tested engine. The registered signal was used to perform the analysis of work uniformity. For the defined operating points, a statistical analysis was performed and the average value of the maximum cylinder pressure was calculated. In

addition, based on the calculated p_{max} values, return maps for the defined operating points were developed.

2. Research object and test stand

The tests were performed using a test bench located in the laboratory of the Lublin University of Technology. The basic element of test stand is the Electromyron EMX 100/10000 electric brake. Table 1 shows the basic brake data. The engine was connected to the brake by means of a shaft and an overload clutch. The test object was two-stroke opposed piston diesel engine.

Table 1. Technical data of the EMX-100 brake

Brake type	EMX-100/10000
Maximum power	100 kW
Maximum speed	10000 rpm
Maximum torque	240 Nm
Brake mass	250 kg
Turning direction	any
Length of the measuring arm	0.370 m

This engine is a development version of a new construction designed for propelling ultra-light aircraft. The engine is called PZL 100 and can generate 100 kW of power. The basic design and functional assumptions for the engine are summarized in Table 2. The engine in the development version is shown in Fig. 1.

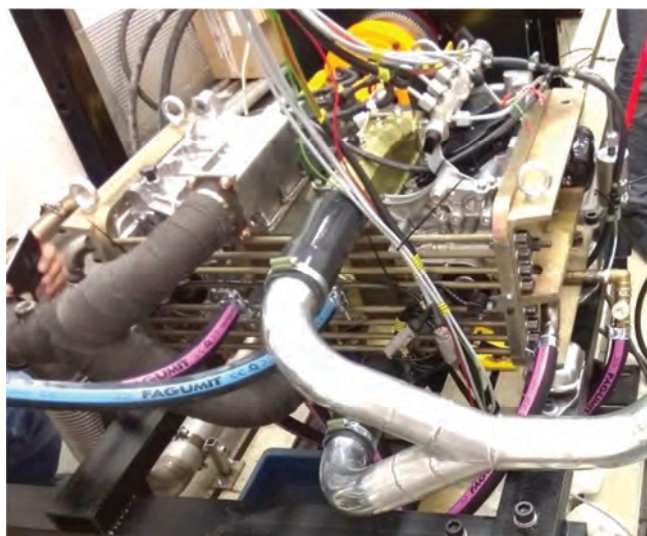


Fig. 1. PZL 100 engine on the test bench

Table 2. Basic parameters of the research object

Take off power	100 kW
Engine speed	4200 rpm
Number of cylinders	3
Bore	65.5 mm
Stroke	72 mm
Compression ratio	22:1
Scavenging	Uniflow

As part of the tests, the pressure in the combustion chamber of the first cylinder was measured. The engine has been equipped with a pressure measuring system located in the first cylinder. For this purpose, a mounting socket was

made in the engine block and cylinder liner in accordance with the pressure sensor manufacturer's instructions. The GH14D sensor from AVL was used to measure the pressure. The sensor is shown in Fig. 2. It is a piezo-electric sensor with a measuring range of 0–25 MPa. Signal conditioning was performed with the AVL MICRO IFEM piezo 4CH SDC amplifier.



Fig. 2. GH14D pressure sensor

In addition, a program for controlling the fuel injection process was developed (using injectors and valves controlling the fuel pressure). The window of the developed program is shown in Fig. 3. The engine is equipped with two injectors per cylinder. This solution reduces the time of fuel injection and enables precise distribution of fuel in the combustion chamber. The PZL 100 engine control system was made at Expansion Chassis from National Instruments. It is a device designed for controlling and collecting data, based on the Zynq-7020 FPGA chip. The platform used has a built-in network interface that allows remote communication with the device. For testing the engine, a program was developed in the LabView software from National Instruments. Due to the need for precise control of injectors and synchronization of the moment of injection occurrence with the position of the crankshaft, the program had to consist of two modules. The first module was responsible for the synchronization and communication of the measuring card with the executive card. The second module was written in the LabVIEW RT software, which is used to create applications that work in real time.

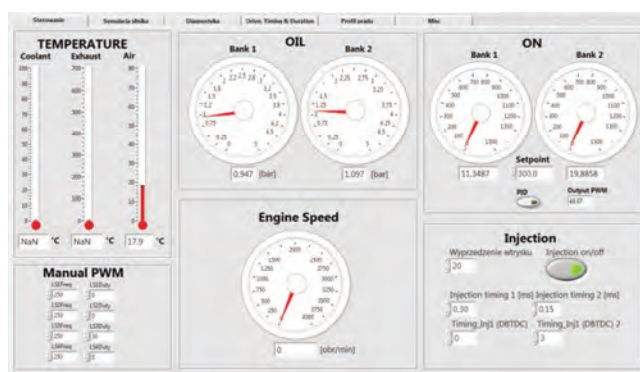


Fig. 3. The engine control panel window

The developed program allowed to set parameters such as:

- oil pressure in fuel rails,
- advance injection in relation to TDC (expressed in crank angle),

- duration of the first injection,
- duration of the second injection,
- offset of the next injection in relation to the previous one.

The results were recorded using the National Instruments 9215 measuring card. A program was also developed in the LabView environment. During the tests, a sampling and recording frequency of 100,000 Hz was adopted. The indicated pressure in the first cylinder and the crankshaft position signal were recorded.

3. Experimental results

The conducted tests were preliminary tests performed after the first start of a new engine type. The purpose of the research was to conduct a basic analysis of the unevenness of engine work. Therefore, the tests were carried out at low values of the rotational speed and the load determined by the torque. For the defined speed value, the power generated was increased by increasing the fuel dose. This is confirmed by the value of the air–fuel equivalence ratio λ shown in Table 3. The lambda value was measured in the exhaust system.

Table 3. Defined engine operating parameters

No.	N [rpm]	M [Nm]	λ [–]
1	1500	11	4,5
2	1500	22	3,8
3	1500	31	2,9

Due to the interference, the signal was subjected to filtration. A low-pass filter was used that removed the frequency bands below 15,000 Hz. The result of this process is presented in Fig. 4. Then the signal was converted using a factor of 25.00758 bar/V. The time of one cycle for the analyzed operating points was 40 ms at the engine speed equal to 1500 rpm. During engine operation for the fuel injection that initiated the combustion, the injector opened at 0.4 ms (20° before TDC). The main injection was given 4° before TDC, and the injector opened at 0.5 ms, 0.7 ms, 0.9 ms, respectively for three measuring points according to Table 3. The obtained $p_{\max}$ values were subjected to statistical analysis, specifying average values and standard deviations.

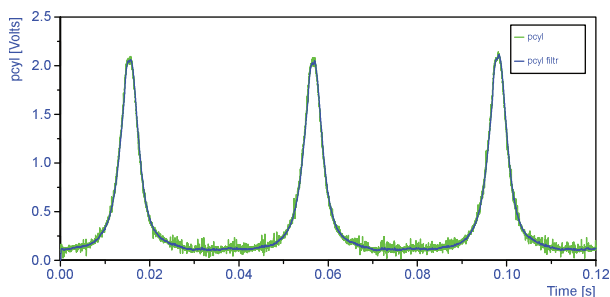


Fig. 4. The signal from the pressure sensor in the first cylinder

Figures 5–7 show cylinder pressure as a function of time for 12 consecutive work cycles for defined operating points. The maximum pressure in the cylinder that was registered was equal to 57.19 bar. Analyzing consecutive

maxima for the three operating points, a significant pressure fluctuation can be observed.

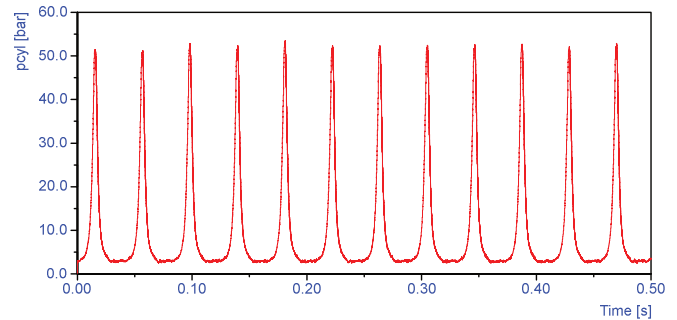


Fig. 5. Fragment of the pressure in the first cylinder – operating point No. 1

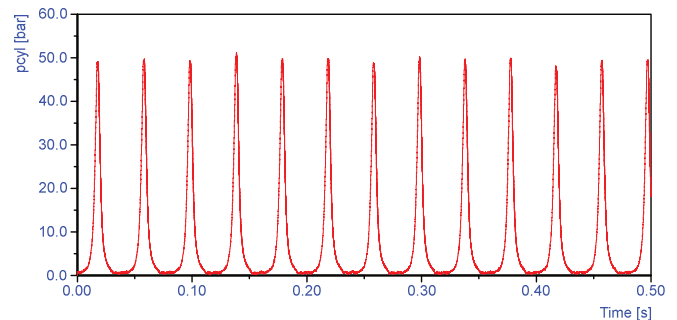


Fig. 6. Fragment of the pressure in the first cylinder – operating point No. 2

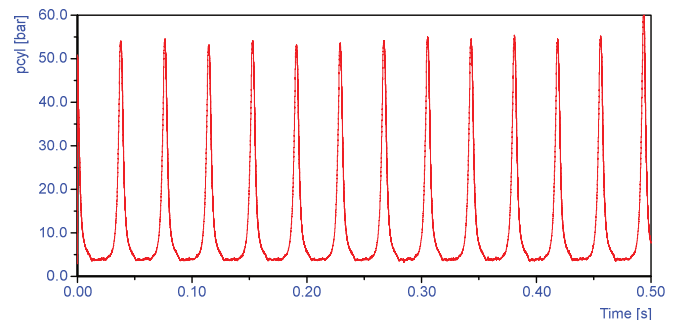


Fig. 7. Fragment of the pressure in the first cylinder – operating point No. 3

Figures 8–10 show calculated maximum cylinder pressure as a function of time for defined operating points. The time of recording pressure signal was 32 seconds for all analyzed work points.

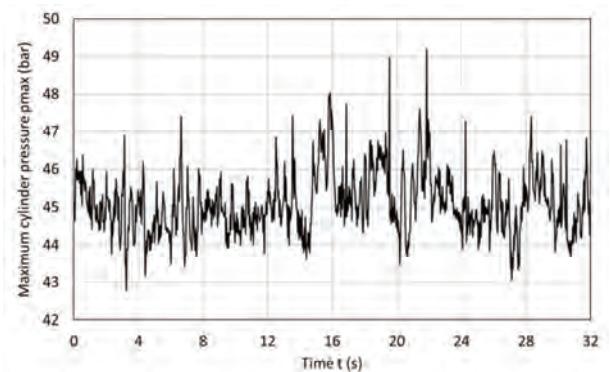


Fig. 8. Maximum cylinder pressure for operating point No. 1

Even in the case of the smallest load there was a significant fluctuation in cylinder pressure. For this working point its value varied from 42.78 to 49.18 bar.

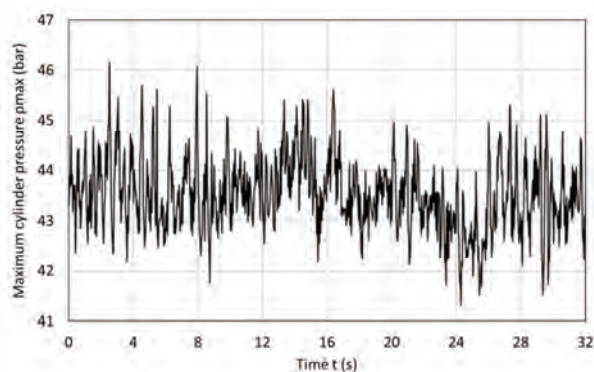


Fig. 9. Maximum cylinder pressure for operating point No. 2

The increase in the load resulted in the reduction of pressure instability in the cylinder. Its value for the 2nd operating point varied from 41.32 to 46.16 bar. The average value of $p_{\max}$ for this case was less than the value corresponding to the 1st operating point.

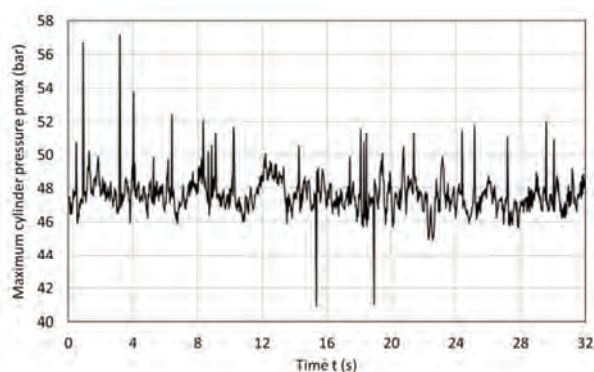


Fig. 10. Maximum cylinder pressure for operating point No. 3

Further increase in the load resulted in the occurrence of single high pressure increases. For example, the average value of $p_{\max}$ for this case was 47.64 bar, and the maximum jump was 57.19 bar. The value of the maximum pressure in the cylinder changed in the range of 40.95–57.19 bar.

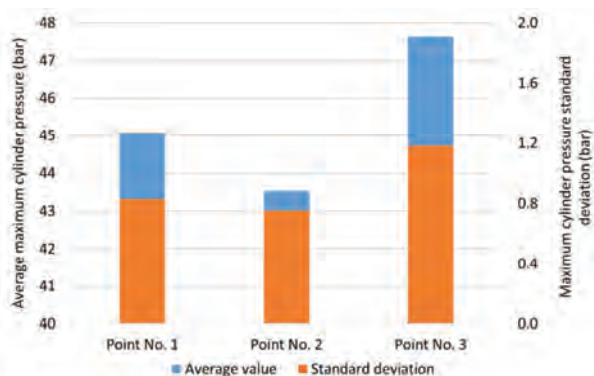


Fig. 11. Average value and standard deviation of maximum cylinder pressure

Figure 11 shows the average value of maximum cylinder pressure and standard deviation of maximum cylinder pressure for analyzed operating points. Due to unstable operation, the increase in load did not result in an increase in the average maximum cylinder pressure. For operating point No. 2, a lower value of $p_{\max}$ was observed compared to point No. 1. In the case of the third operating point, the highest value of standard deviation occurred. This means a large variation of pressure during the measurement.

Based on the results of the pressure measurement, return maps for the tested engine were developed. The return map shows the relationship between the given work cycle and the next cycle for the selected engine operation parameter.

Figure 12 shows the return maps for the maximum cylinder pressure for the considered operating points.

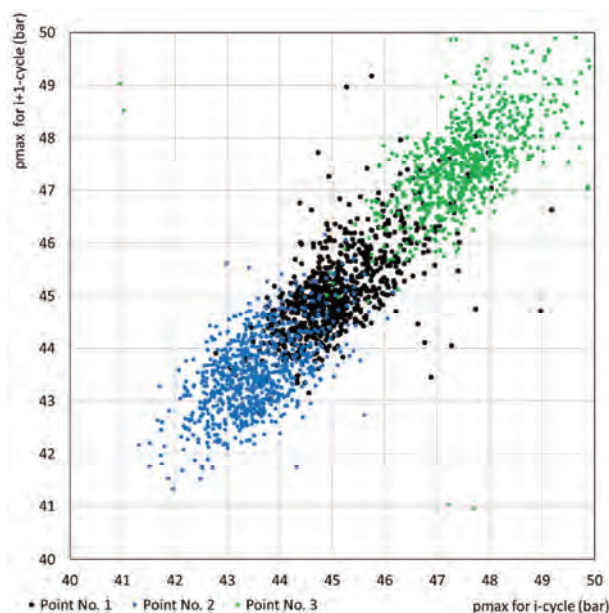


Fig. 12. $p_{\max}$ return maps for the analyzed operating points

For all the work points considered, there was a significant dispersion of the measurement points. This means that the engine operation was not stable under given conditions. The distribution of points along the line confirms that the maximum pressure is unstable. At the same time, it can be concluded that the subsequent values depend on the previous values.

4. Conclusions

The paper presents experimental results of two-stroke opposed piston engine tests. The measurements performed on the test bench made it possible to register the pressure in one of the engine cylinders.

The recorded signal was subjected to filtration. Maximum pressure values were obtained for the analyzed operating points, which were subjected to statistical analysis. In addition, return maps for $p_{\max}$ were developed.

A significant fluctuation in cylinder pressure was observed, which translated into unstable operation. The measured pressure for analyzed operating points ranged from 41 to 57 bar.

As part of further work, optimization of the injection process and control algorithm is planned.

Acknowledgements

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Nomenclature

CI	compression ignition	EGR	exhaust gas recirculation
CPBC	cylinder pressure-based control system	SI	spark ignition

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Analysis of the macrostructure of the fuel spray atomized with marine engine injector

One of the main problem influencing the combustion process in the cylinder of the marine engine is an fuel spray phenomena. The parameters describing the shape of the fuel spray are named macro parameters. This article presents the research results of the macrostructure parameters of the fuel spray atomized with the marine engine injector. The research were carried out by optical visualization measurement method of Mie scattering. The diameter of nozzle injector was 0.375 mm and L/D coefficient 8.3. In these research were considered different parameters of injection opening pressures and backpressures in the constant volume chamber. Generally conclusions are: the opening pressure influence on maximum spray tip penetration, spray cone angle and influence on speed of the injected fuel. The increase of backpressure into the constant volume chamber causes the reduction of spray tip penetration and the increase of the spray cone angle.

Key words: marine diesel engine, marine fuel injector, spray tip penetration, spray cone angle, opening pressure, backpressure

1. Introduction

Diesel engines are the main type of propulsion of marine vessels. Typically, there are low-speed two-stroke diesel engines and medium-speed four strokes. Burning diesel oil by marine engines effects the emission of toxic compounds into the atmosphere. The amount of toxic compounds in exhaust gas of marine diesel engines are regulated by International Maritime Organization (IMO) [1]. Reduction of emission of toxic compounds by marine diesel engines is one of the most important aspects in nowadays diesel engines researches and engineers. The maximum engine efficiency and low emission toxic compounds are main aim during at the design stage of marine diesel engines.

The process of combustion in a marine diesel engine cylinders are mainly determined by fuel injection process [2].

High pressure of injection and controlled fuel delivery are the advantages of the common rail system. For this purpose, the common rail injection system is used in marine diesel engines. The first Common Rail system in marine, medium speed, 4 stroke engine was installed in 2001. High injection pressure are preferred for reduce excessive emission of toxic compounds [3]. Therefore, it is important to researches conducted the studies of influence injection pressures on the fuel spray and combustion process into cylinders. The combustion process in the cylinder of the marine diesel engines is diversified. The process of fuel delivery to cylinder, fuel injection and fuel combustion should be analyzed separately. Furthermore, the research of the phenomenon occurring in the combustion chamber are fast changing. The subject of the injection fuel process into the real operating marine engines may belong impossible or very expensive. There is alternative such is: the constant volume chamber – simulating conditions of combustion chamber marine diesel engines, optical methods – measurements of the injection process into combustion chamber [4].

Fuel delivered by marine engine injector to cylinder is atomized, evaporated and mixed with air. Injected fuel had a shape of a cone with a spherical forehead described with macro parameters: tip penetration and spray cone angle. Tip penetration and spray cone angle are present fuel droplets

ability to penetrate in gaseous environment in combustion chamber. Over the years, many researches have performed a lot of studies in order to research the spray tip penetration and spray cone angle. In the '80s and '90s the pioneering works about research of macro parameters were presented by Hiroyasu & Arai [5] and Naber & Siebers [6]. Author's proposed the diesel fuel atomization models which assumes that spray tip penetration is linear proportional to the time only at the early injection period. After this time, the spray tip penetration evaluated with proportional rate to square root of time. The Hiroyasu & Arai or Naber & Siebers fuel atomization model are basis for researches to development innovative models. Macro parameters are mainly depend for geometrical parameters of fuel injection system and environmental conditions in cylinder. Generally, the geometrical and operating parameters are [7]: diameter of nozzle, L/D ratio, energizing time of fuel injection, injection pressure and others. The parameters of environmental conditions are [8, 9]: air swirls movement, backpressure or temperature of fuel and air.

Payri R. et al. [10, 11] have performed researches on the influence of orifice geometry on the flow at the nozzle exit and its effect on the macro parameters. The conclusion is that cylindrical nozzle provided the highest values of liquid spray tip penetration at low injection pressures in range 30 and 80 MPa. The change temperature during of combustion process in cylinder causes changes of the fuel density and viscosity. Such a phenomenon caused the poor atomization [8] and vaporization [12, 13]. In the literature, there are many studies concerning the macro parameters of the fuel injection depending on injection pressures and backpressures. Du W. et al. [14] presented the influence injection pressure for behaviors of liquid phase spray and vapor spray. The measurement were performed in the constant volume chamber. High speed camera and Schlieren visualization set-up are used to registration results. Their test results presented influence injection pressure for spray penetration rate and the effect of pressure on general development of spray cone angle is negligible.

Presented studies of the macro parameters of diesel fuel spray are presented for fuel injector of the lower of geomet-

rical parameters compared the geometrical parameters of marine diesel engines. Therefore, study of diesel spray parameters of marine engines is needed.

The paper presents results of macroscopic parameters of fuel spray for the chosen backpressure conditions and chosen opening pressures of marine fuel injector.

2. Laboratory setup

Laboratory studies presented in this article were carried out in Warsaw University of Technology. The chosen parameters correspond with the parameters observed in marine engines. The nozzle diameter was of 0.375 mm and L/D coefficient 8.3 respectively. The all measurements of fuel spray were conducted of the ambient temperature. The temporary evolution sprays parameters was observed in the constant volume chamber (Fig. 1, Table 1). The environmental conditions in constant volume chamber was backpressure 3.2 MPa and 4.3 MPa. Those backpressures were defined adequately operation marine diesel engine with to half- and full- load engine operation.



Fig. 1. Constant volume chamber

Marine fuel injector was installed at the top in the constant volume chamber (Fig. 2). This configuration allow to observation of single jet evolution fuel sprays. The observation area of fuel sprays equals up to 100mm (half of the cylinder bore).

Table 1. The parameters of the constant volume chamber

Dimensions	200 × 200	mm
Optical view	100	mm
Backpressures	3.2, 4.3	MPa
Measurement space	nitrogen	–
Number observation windows	3 (all 4)	–

Fuel injector used in this study is the part of laboratory marine diesel engine Sulzer 3 Al 25/30. The injector is a conventional pressure opened diesel injector equipped with UPS (Unit pump system). Fuel injector supplies diesel oil of 300 K temperature. The tested spray of diesel was made by one active injector nozzle. Others injector nozzle was plugged. In the presents study the injector nozzle with

0.375 mm diameter and 8.3 L/D coefficient was investigated. The fuel was supplied to the injector by a high pressure common rail system equipped with fast-acting electromagnetic valve. The pressure behind the injector was measured by Kistler type 4067E piezoresistive pressure sensor. The specifications of the pressure sensor was showed in [15]. The fuel pressure in system equals 50 MPa. The different opening pressure of marine fuel injector was considered in this study. The opening pressure range was 15 MPa, 25 MPa and 35 MPa, respectively. Measured diesel oil viscosity and density at 40°C was of 2.35 mPa·s and 816.1 kg/m³ respectively. The temperature of test was 300 K. In this work, the optical Mie scattering method was used [4, 16]. The experimental setup system is shown in Fig. 2.

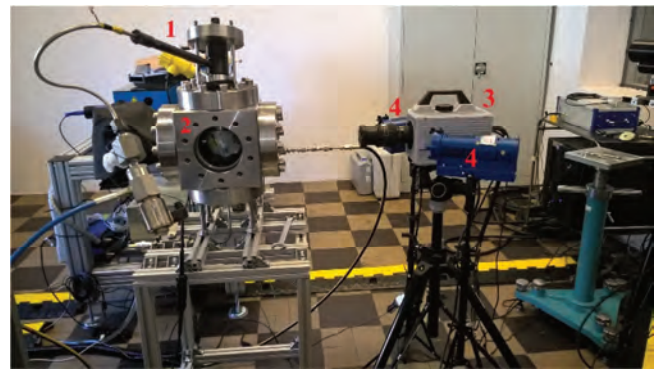


Fig. 2. The experimental setup system: 1 – marine engine injector, 2 – constant volume chamber, 3 – Photron SA1.1 high-speed camera, 4 – 2x halogen lights 0.5 kW

In Mie scattering method, important aspect is adequate lighting of sheet spray. On the based measurements requirements of macro spray parameters were chosen integral illumination of the spray.

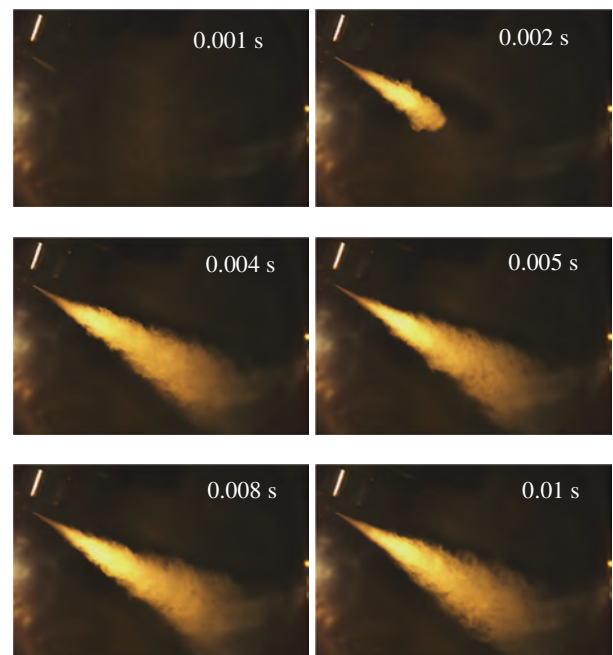


Fig. 3. The chosen pictures of diesel spray in time – the nozzle diameter 0.375 mm, L/D 8.3, opening pressure 25 MPa and backpressure 4.3 MPa

Two halogen lights were used. In the determination of global spray parameters, the integral type of illumination is very well. The diesel fuel sprays were observed by a Photron SA1.1 high speed camera. The images were recorded at a frequency of 40 kHz and resolution of the recorded images was 512×256 pixels.

The measurement was repeated three times at each point. In single measurement was generated about 500 images, where about 250 images has been used to photo processing. The Fig. 3 presented chosen pictures generated from the marine fuel injector in the constant volume chamber. Selected photos contained the range injection process from the beginning of the injection to the complete development of the spray. Prior to the generated of the macro parameters of the diesel spray, images had to be prepared. The first step of photo processing was scaling from pixels to millimeters. At the same time, it had to be verified or improved of the nozzle position in the photo. One pixel is equal 0.13 mm in this photo processing. The next step was used spray isolator function to separate the spray from the ambient and to eliminate reflections and image noises within spray. The example of images processing effect is presented in Fig. 4. The mentioned photo processing were carried out by DaVis v8.4 software.

3. Results and discussion

The Fig. 4 presents example results of macro parameters of the tested diesel spray. The spray tip penetration (STP) is defined as a maximum distance of fuel spray from the hole nozzle outlet [17]. According to the presented results in Fig. 4, spray cone angle is defined by two lines on the boundaries on the left and right sides which made cone.

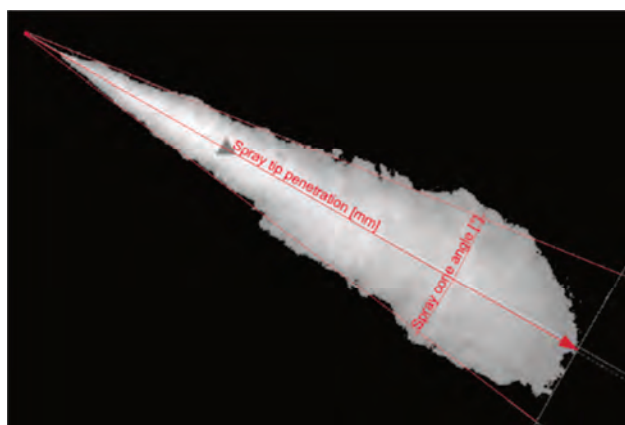


Fig. 4. Macro spray parameters

3.1. Spray tip

Figure 5 presents STP for opening pressures of 15, 25 and 35 MPa for two backpressures in the constant volume chamber of 3.2 MPa and 4.3 MPa in non-evaporating conditions. The test results are the arithmetic average of the three repetitions of the measurement. All presented cases are similar to studies by Hiroyasu&Arai [5]. The spray tip penetration is presented in range to the time of 0.01 s, because after 0.008 s it achieves the constant level of maximum penetration. The maximum spray penetration is range 70–100 mm for all the considered opening pressures, backpressures. In the early stage of diesel fuel injection

STP increases with time, until achieving a stabilization stage. All the considered curves in time of STP may be defined in two stages: the initial increase stage and the steady development stage. During the steady development stage STP does not increase but only oscillates about its average range. According to results presented in [18], when the distance of the spray tip penetration increases, droplets diameters decrease. This is caused by primary and secondary breakup of droplets of fuel spray and aerodynamics resistances in the constant volume chamber. The steady development stage of STP presents of maximum of breakup droplets. This phenomena caused that STP did not grow up. The start time of stabilization stage of STP was presented by the vertical lines on the Fig. 5.

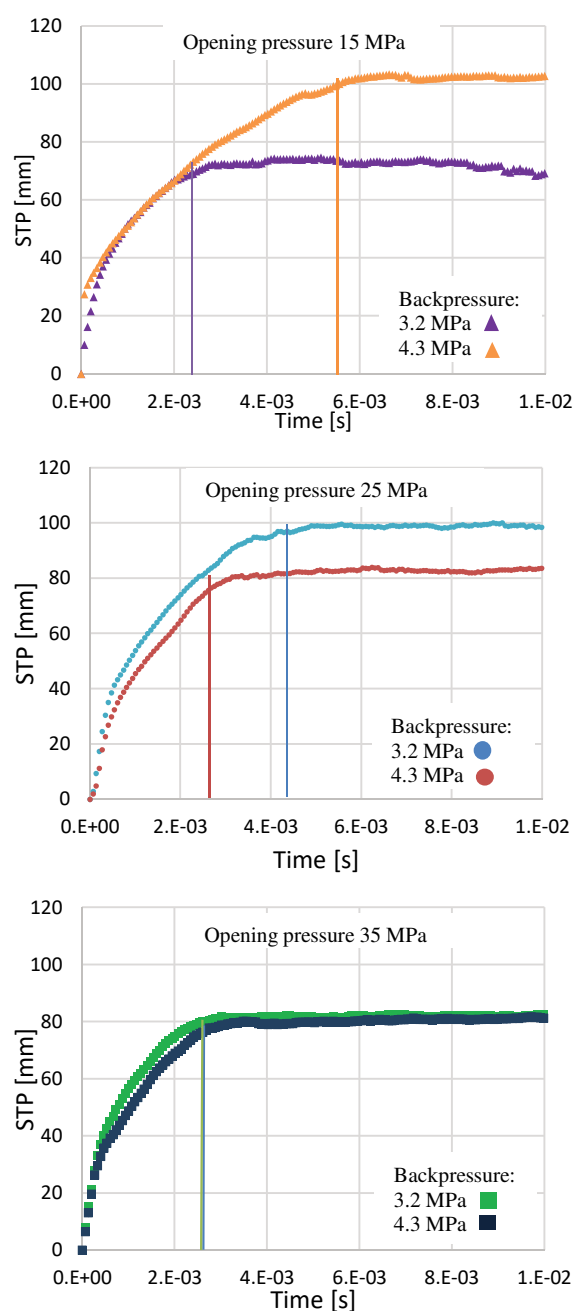


Fig. 5. Spray tip penetration for 0.375 mm diameter nozzle, opening pressures: a. 15 MPa, b. 25 MPa, c. 35 MPa, backpressures: 3.2 MPa and 4.3 MPa

The initial penetration STP is determined by the injection dynamics, instead of the ambient conditions [19]. This can be seen on the Fig. 5 where the increase of backpressure has little effect on STP at the initial increase stage where the diesel spray is dense. This may be due to the much higher injection pressure compared to the backpressure in constant volume chamber. Higher opening pressure can increase kinetic energy of diesel fuel spray. The STP of fuel diesel has larger kinetic energy to penetrate and results in a faster achieving stabilization stage and highest of STP.

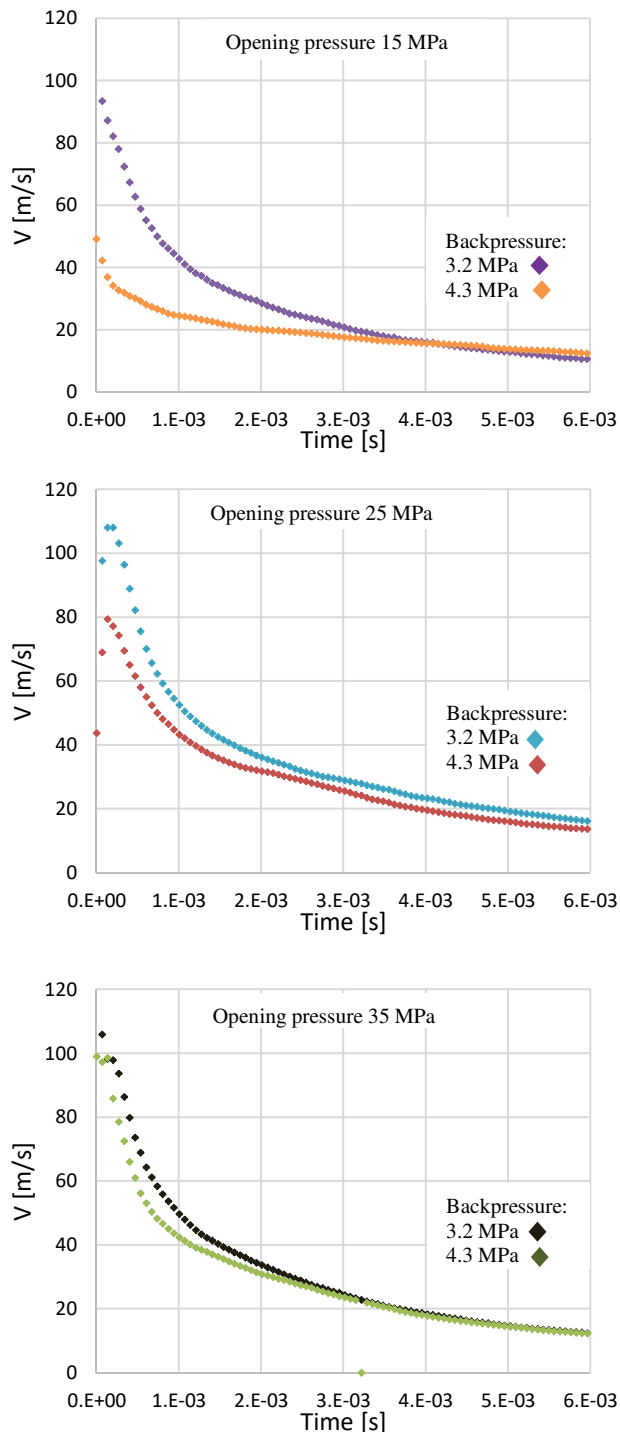


Fig. 6. Speed of STP: different opening pressures and backpressures: a. 3.2 MPa and b. 4.3 MPa

After initial propagation in time of STP, the spray becomes thin and the effect of aerodynamic force gradually increases. The kinetic energy of diesel fuel spray was reduced. It can be explained that the droplets of fuel in STP were slowed by increases of aerodynamic resistances. The results presented on Fig. 5 is expected for opening pressure of 25 MPa. Increase of backpressure of 3.2 MPa to 4.3 MPa causes significant reducing of STP and reducing time of achieving of stabilization stage of STP for opening pressure of 25 MPa. For higher opening pressure (35 MPa) the change is not observed. The opposite situation was observed for opening pressure 15 MPa: higher backpressure caused elongation in time achieving and increases the stabilization stage of STP.

On the other hand, opening pressure and backpressure have influenced the speed of STP in initial stage. On the Fig. 6 presents curves of speed V [m/s] of STP in time for considered opening pressures of marine diesel injector and backpressures in the constant volume chamber are presented. The increase of opening pressure was caused the increase of fuel injection speed at the initial stage of injection.

According to the results, for increase backpressure of 4.3 MPa there is observed reduction of initial speed of STP in comparison to backpressure of 3.2 MPa for all considered opening pressures. The biggest influence of backpressure on initial speed of STP is for opening pressure of 15 MPa. For opening pressure 25 MPa and 35 MPa influence of backpressure caused small difference in initial speed of STP.

On the Fig. 7 presents averages STP of stabilization stages. Increase of opening pressures caused increases of STP for backpressure 3.2 MPa. Opening pressure of 25 MPa caused the increase of STP by approx 34% and opening pressure of 35 MPa by approx 13% in compared to STP for lower opening pressure.

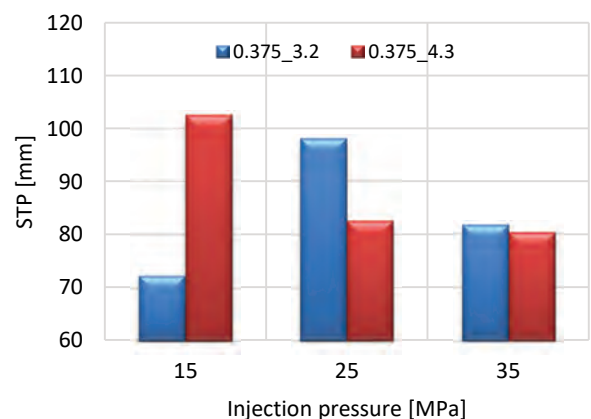


Fig. 7. The averages STP of stabilization stages for backpressures 3.2 MPa and 4.3 MPa

According to results presents on Fig. 7, the STP decreases with increases of opening pressures, for backpressure of 4.3 MPa compared to backpressure of 3.2 MPa. For opening pressure of 15 MPa the STP is the highest. The change of opening pressure from of 15 MPa to 25 MPa caused a reduction of STP of about 20 %. However, between the opening pressures of 25 and 35 MPa, 2% difference of STP was observed.

In conclusion, opening pressure has effect on STP of marine diesel engine injector. The course in time of STP is created by injection dynamics of STP and minimal effect of backpressure in constant volume chamber during initial stage. The backpressure in stabilization stage has the main influence on maximum STP.

3.2. Spray cone angle

Figure 8 presents the course of the spray cone angle (SCA) in time for opening pressures of 15, 25 and 35 MPa and backpressures of 3.2 MPa and 4.3 MPa in non-evaporating conditions.

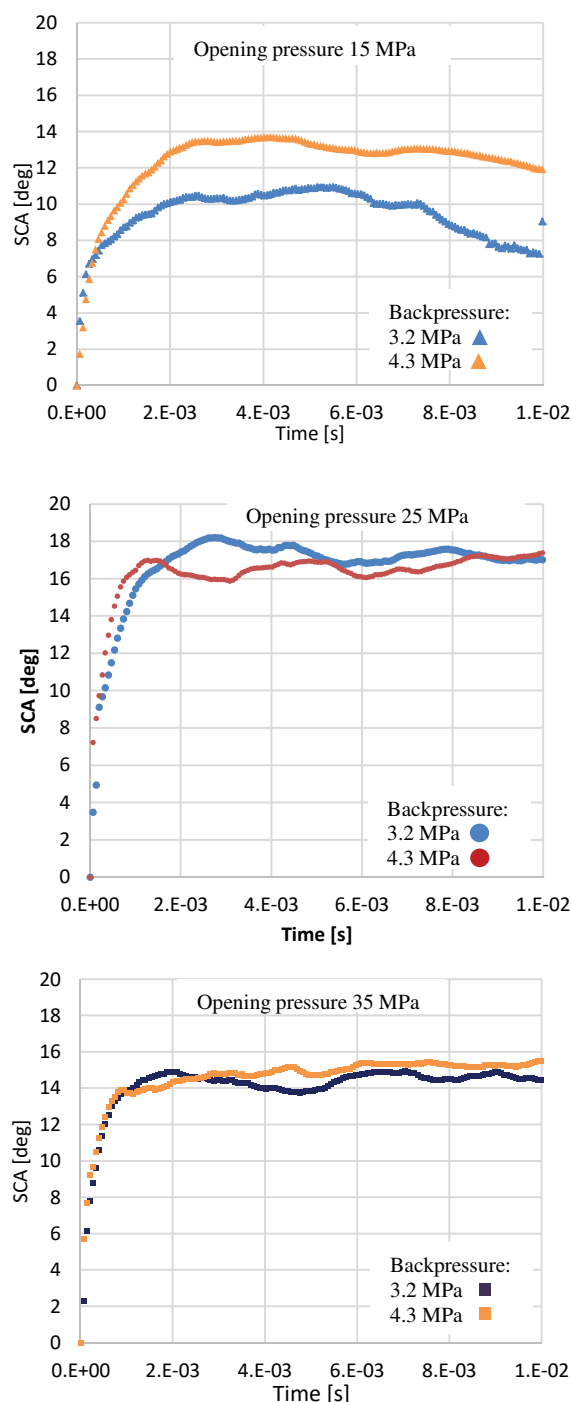


Fig. 8. Spray cone angle for 0.375 mm diameter nozzle, different opening pressures and backpressures: a. 3.2 MPa and b. 4.3 MPa

All the presented characteristics of SCA appeared the initial stage. This initial stage lasts approximately 0.001s in each considered cases. During initial stage of SCA, the rate of SCA is not the same. The rate of SCA depending on considered backpressure are a little differences for opening pressures of 15 MPa and 35 MPa. For opening pressure of 25 MPa increase rate of SCA for backpressure 4.3 MPa in compared backpressure 3.2 MPa was observed. This phenomenon can be attributed to the ambient gas resistance generated in the constant volume chamber which influenced on the surface wave around the periphery of spray. The initial stage is followed by a period with relatively constant “quasi-steady” spray angle. The presented results are similar to [6]. The SCA during the initial transient stage can significantly exceed the quasi-steady angle or can remain on the same level. The SCA in quasi-steady spray angle for the marine diesel injector is 10° to 18°.

Based on Fig. 9, for all considered backpressures, the SCA increases along with the increase in the opening pressure. This is consistent with the research results presented in [5]. In the presented research result the maximum SCA was achieved for opening pressure of 25 MPa in both considered backpressures in the constant volume chamber.

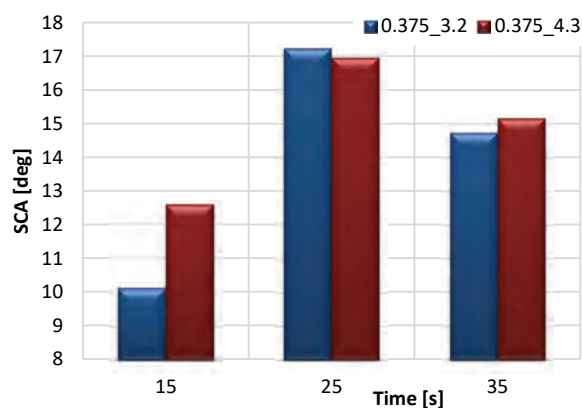


Fig. 9. Effect of injection pressure on maximum SCA for backpressures 3.2 MPa and 4.3 MPa

Between opening pressures of 15 MPa and 25 MPa, the biggest difference in SCA is observed. The mentioned difference of SCA is 4°–7° while between opening pressures of 25 MPa and 35 MPa difference is about 2°. This is due to the larger nozzle in marine diesel engines which changes the outlet flow pattern in compared small general diesel engines. In marine diesel engines with relatively big fuel nozzles, the fuel spray is along the axis of nozzle while in diesel engines fuel is spray divergently [20].

In case of the increase of backpressure from 3.2 MPa to 4.3 MPa in the constant volume chamber, there are observed significant increases of SCA for of 15 MPa opening pressure (Fig. 9). For other considered opening pressures, influence change backpressures on SCA is insignificant. Increasing opening pressure can change the kinetic energy of the fuel which causes friction between fuel and ambient gas resulting in breaking up. Therefore, increases of backpressure cause increases of SCA for marine engine injector.

4. Conclusions

This paper presents tests of macro parameters of fuel diesel spray from a marine engine injector. According to presented research the following conclusions may be formulated:

- The opening pressure of marine engine injector influences the maximum spray tip penetration and the spray cone angle;
- The opening pressure of marine engine injector influences the initial speed of spray tip penetration. Together with the increasing of opening pressures of marine injector the initial speed of fuel spray also increases; During steady development stage of spray tip penetration, the influence of opening pressure can be neglected for higher opening pressures;

- Backpressures in constant volume chamber influence macro parameters of diesel fuel spray in steady development stage. The increase of backpressure in the constant volume chamber causes the reduction of speed of spray tip penetration, maximum of spray tip penetration and the spray cone angle.

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Nomenclature

STP Spray Tip Penetration

SCA Spray Cone Angle

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The impact of ignition system damage on the emission of toxic substances in a spark-ignition internal combustion engine

The article discusses the impact of ignition system damage on the emission of toxic subcategories in a spark-ignition internal combustion engine. The aim of the work was to develop an analytical model of ignition system diagnostics, test performance and comparative analysis of the results of simulations and experiments. The model developed allows to analyse the basic parameters of the ignition system affecting the content of toxic substances in the exhaust. Experimental tests were carried out using the MAHA MGT5 exhaust gas analyser for four different combustion engines fueled with petrol at various operating conditions. During the tests, the content of toxic substances in the exhaust gas of a properly working engine and the engine working with damage to the ignition system were registered. The tests will be used to assess the impact of the damage of the spark-ignition engine on the emission of individual components of toxic fumes.

Key words: spark-ignition internal combustion engine, emission of toxic substances, exhaust standards, ignition system defects, diagnostics

1. Introduction

The unfavourable influence of the automotive industry and road transport on the state of the environment is felt in Poland as one of the pressing problems to be solved. The ever more modern efficiency and ecology solutions introduced in vehicles are offset by the constantly increasing number of vehicles on the road, and the amount of toxic exhaust components released into the atmosphere [9].

Protection of the natural environment, carried out for a long time, has gained particular importance in recent years. The two basic reasons behind it are the rapid, uncontrolled progress in the development of the automotive industry and the distribution of emission sources in urban and industrial areas.

Regardless of the type of engines driving modern vehicles, three fundamental factors are taken into account in their evaluation [1, 11]:

- fuel consumption,
- content of toxic substances,
- engine flexibility.

Meeting the consequent conditions is a challenge requiring significant financial outlays to carry out research on solutions in manufactured vehicles. It is assumed that the greatest environmental impact in a vehicle in terms of harmfulness comes from its engine. It is impossible to develop an engine control programme that would ensure zero emissions.

Exhaust emissions tests in terms of toxicity are performed on new engines to meet the approval conditions in accordance with standards that have varying global emission limits in individual years. European emission limits are defined by Euro 1-6 standards for cars and trucks. These standards concern the emission of four basic exhaust gas components: CO, CO₂, HC, NO_x and PM particulate matter. Analysis of spark-ignition internal combustion engines requires a detailed examination from the point of view of the impact of the ignition system parameters on exhaust

emissions. The ignition parameters affecting the emission of toxic substances include the ignition advance angle, the spark discharge energy and the distance between the spark electrodes [3, 8, 10].

2. Analysis of the impact of ignition system parameters on the emission of toxic substances

Depending on the composition of the fuel and air mixture, the change in the ignition advance angle affects the temperature of the combustion process. The maximum engine power is obtained at the highest temperatures, providing a rich mixture to the combustion chamber. In the case of poor mixtures it is necessary to increase the ignition advance angle, which is associated with a change in the content of carbon monoxide, hydrocarbons and nitrogen oxides in the exhaust.

2.1. Ignition advance angle

By making changes of the ignition advance angle in relation to the optimum value, the pressure and temperature in the combustion chamber are reduced, which reduces the amount of hydrocarbons. This is also connected with the increase of temperature during the expansion and discharge stroke, which causes the combustion of toxic substances in the exhaust system after the combustion process [12].

Ignition timing retardation reduces the amount of nitrogen oxides in the exhaust fumes for lean and stoichiometric mixtures. This is due to the lower combustion temperature, which is the result of a partial transfer of the combustion process to the expansion stroke.

2.2. Energy of the spark discharge

The energy of the spark discharge has a large impact on the emission of toxic substances, which should be greater than the energy of ignition for a given fuel. The discharge energy is the product of the voltage on the spark plug, the intensity of the discharge current and the duration of the discharge.

In the case of rich mixtures ($\lambda < 1$), the current intensity and the spark discharge time have no effect on the emission of hydrocarbons, nitrogen oxides and unit fuel consumption. When using a lean mixture ($\lambda > 1$) it is necessary to generate a higher spark current and a longer discharge time [2, 7].

2.3. The distance between the candle electrodes

The emission of toxic substances depends on the way the flame front propagates in the cylinder chamber, the location of the spark plug in the head and the distance between the electrodes. The position of the candle close to the cylinder wall increases the hydrocarbon emission associated with the flame extinguishing effect. A very similar phenomenon occurs when the diameter of the centre electrode and the gap between the electrodes increase.

3. Model of the high voltage course in the ignition system

The development of the voltage waveform model involves the use of simplifications. Assumptions are made about its linearity, and sometimes also stationarity. In stationary objects, the reaction depends only on the input signal and there are no changes in their properties due to the passage of time.

The development of an analytical model of a high voltage course requires taking into account all parameters and determining the mutual relationships between them. The obtained equations describe the dependencies between particular sizes of the ignition system.

The electric scheme of the ignition system is shown and described in Fig. 1 [6].

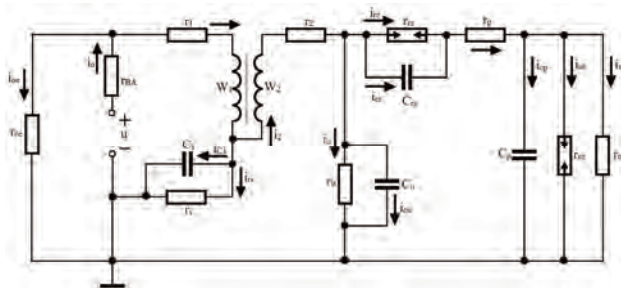


Fig. 1. Electrical diagram of the ignition system

where: r_{oe} – resistance of electric energy receivers, r_{BA} – internal resistance of the battery, $r_1 = r_{war} + r_{wl}$ – total resistance of the primary winding of the ignition coil, r_s – resistance in the connecting circuit, $r_z = r_{wz} + r_w$ – resistance of the secondary coil of the ignition coil, r_{iz} – insulation resistance, r_{rz} – ignition distributor resistance, r_p – resistance of high-voltage wires, r_{sz} – resistance of the gap between the spark plug electrodes, r_n – carbon deposit resistance on the spark plug, C_1 – capacitor capacity, C_u – capacity of the secondary winding of the ignition coil, C_{rz} – ignition distributor capacity, $C_p = C_{pw} + C_{sz}$ – the capacity of wires and spark plugs, W_1 – number of turns of the primary winding of the ignition coil, W_2 – number of windings of the secondary coil of the ignition coil, U – battery voltage.

The electrical diagram of the ignition system shown in Figure 1 is described by the following equations:

$$U = i_o \cdot r_{BA} + i_1 \cdot r_1 + W_1 \cdot S \cdot \frac{dB}{dt} + u_{c1} \quad (1)$$

$$U = i_o \cdot r_{BA} + i_{oe} \cdot r_{oe} \quad (2)$$

$$i_o = i_1 + i_{oe} \quad (3)$$

$$i_1 = i_{c1} + i_{rs} + i_2 \quad (4)$$

$$H \cdot l = i_1 \cdot W_1 + i_2 \cdot W_2 \quad (5)$$

$$H = A \cdot B \quad (6)$$

$$u_{c1} = i_{rs} \cdot r_s \quad (7)$$

$$r_s = \begin{cases} r_{s0} - \text{open circuit} \\ r_{s1} - \text{compact circuit} \end{cases} \quad (8)$$

$$u_{c1} = \frac{1}{C_1} \int i_{c1} \cdot dt + u_{c1}(0) \quad (9)$$

$$W_2 \cdot S \cdot \frac{dB}{dt} + i_2 r_2 + u_{cu} + u_{c1} = 0 \quad (10)$$

$$i_2 = i_{iz} + i_{cu} + i_p \quad (11)$$

$$i_p = i_{rz} + i_{cr} = i_{cp} + i_{sz} + i_n \quad (12)$$

$$u_{cu} = \frac{1}{C_u} \int i_{cu} dt + u_{cu}(0) \quad (13)$$

$$u_{cu} = u_{cr} + i_p \cdot r_p + u_{cp} i_2 = i_{iz} + i_{cu} + i_p \quad (14)$$

$$u_{cu} = i_{iz} \cdot r_{iz} \quad (15)$$

$$u_{cr} = \frac{1}{C_{rz}} \int i_{cr} dt + u_{cr}(0) \quad (16)$$

$$u_{cr} = i_{rz} \cdot r_{rz} \quad (17)$$

$$r_{rz} = \begin{cases} r_{rz0} - \text{without a spark} \\ r_{rz1} - \text{with a spark} \end{cases} \quad (18)$$

$$u_{cp} = \frac{1}{C_p} \int i_{cp} \cdot dt + u_{cp}(0) \quad (19)$$

$$u_{cp} = i_{sz} \cdot r_{sz} = i_n \cdot r_n \quad (20)$$

where: S – cross-sectional area of the ignition coil, B – magnetic induction, t – time, H – intensity of the magnetic field, l – average length of the magnetic field lines, A – constant.

Mathematical equations (1)–(20) describe the changes in the electrical size of the ignition system and the relationships between the various ignition systems.

Using the above mathematical equations and software in the LabView environment, a voltage waveform was generated on the coil's secondary winding shown in Fig. 2.

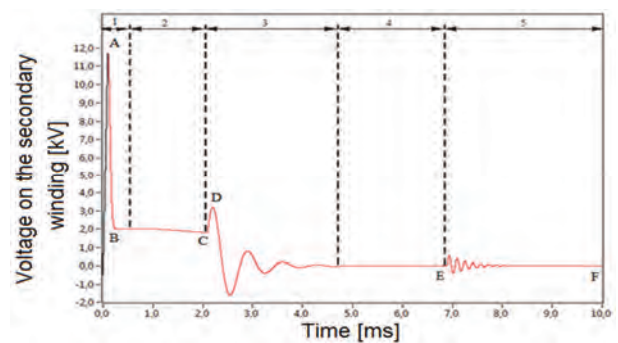


Fig. 2. The ignition signal on the secondary side is made with LabView software

The course of the ignition voltage on the secondary side is made up of three characteristic sections:

- the section of spark operation, which includes the time interval "1" and "2",
- a transition period within the time interval "3" and "4",
- a short circuit which includes the course from the time interval "5".

When developing a model of the ignition voltage course on the secondary side, the following damage to the system was taken into account:

- the ignition voltage drop below 5 kV in the event of a short-circuit in the ignition wire to the mass reflects damage to the spark plug or coil,
- an increase in the ignition voltage above 5 kV in the event of a short-circuit in a single ignition wire reflects damage to the distributor or high-voltage cables,
- an increase in the ignition voltage above 5 kV in the event of shorting all ignition wires reflects damage to the distributor or interruption of the ignition coil high voltage wire,
- increase in the ignition voltage above 12 kV on selected cylinders reflects the increased inter-electrode gap, break in the high voltage circuit or uneven composition of the fuel-air mixture for individual cylinders.

The impact of the above failures on the course of the ignition voltage is shown in Figs 3–6, while the results of toxicity tests are presented in the author's publications [4, 5].

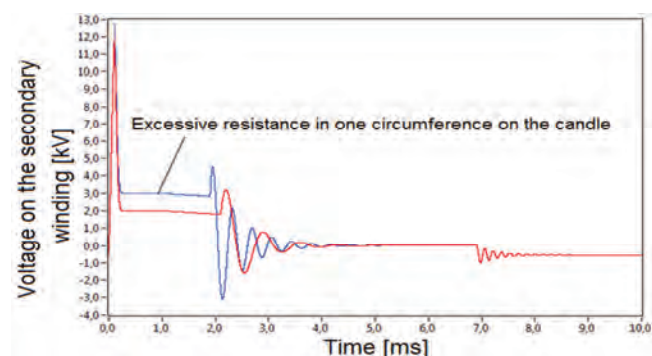


Fig. 3. Changes in the ignition voltage course on the secondary side in the event of shorting a single ignition wire to ground

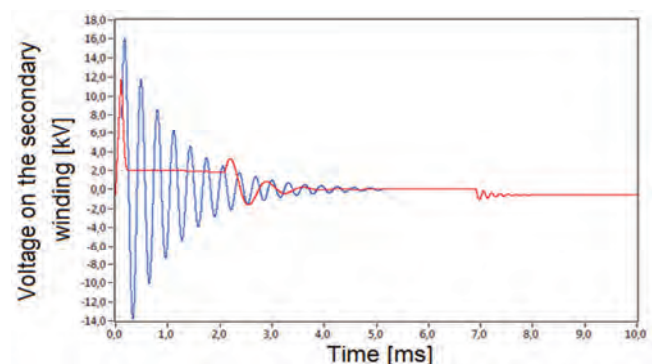


Fig. 4. Changes in the ignition voltage course on the secondary side in the event of a break in the spark plug circuit and no spark

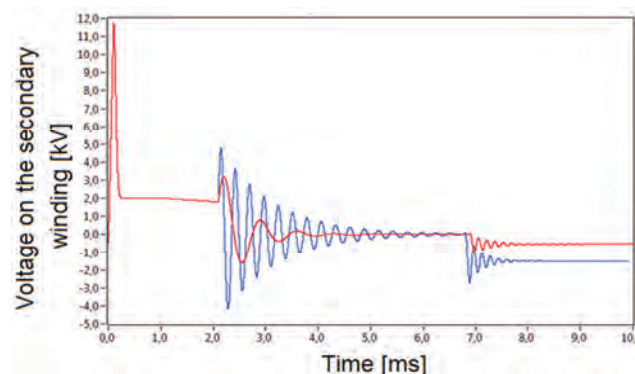


Fig. 5. Changes in the ignition voltage course on the secondary side in the event of a high voltage break in the circumference of the candle

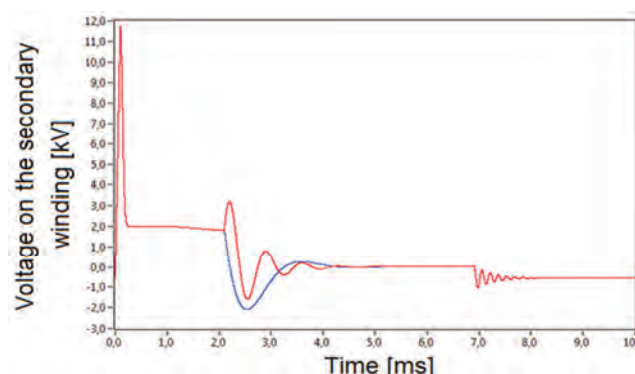


Fig. 6. Changes in the ignition voltage course on the secondary side in the case of shorting of the winding in the ignition coil or piercing of the capacitor

4. Experimental research

Tests were carried out for the BMW M10B18 (Fig. 7) and Fiat 1.1 (Fig. 8) internal combustion engine fueled with petrol. They were equipped with indirect MPI fuel injection and an electronic ignition system. The BMW M10B18 engine have mechanical ignition distributor and has not catalytic converter. The Fiat 1.1 engine has catalytic converter.

The use of FSA, KTS, MOT motor diagnosis allowed to register changes taking place during simulation of ignition system faults. In conjunction with the MAHA MGT5 analyser (Fig. 9), the emission of toxic substances to change parameters in electric circuits was registered.



Fig. 7. Test stand of the BMW M10B18 engine.



Fig. 8. Test stand of the Fiat 1.1 engine



Fig. 9. MAHA MGT 5 exhaust gas analyzer

Ignition system tests including changes in electrical circuit parameters and toxicity included:

- determination of emission of correctly functioning spark-ignition engines,
- determination of changes in the composition of exhaust gases in the event of a lack of continuity in the circuit of one of the high voltage wires,
- to determine the toxicity of exhaust gases if the gap between the spark plug electrodes is reduced or increased,
- determination of toxicity of fumes in the event of a change in the location of the spark plug electrode,
- simulation of the phenomenon of switching off individual cylinders using the MOT 250 diagnoscope,
- control of the secondary circuit of the ignition system (full adaptation) by means of FSA 720.

Prior to the simulation of the ignition system faults, activities were carried out to check the correct operation of the engines and individual toxic components in the exhaust were registered. The gas analyzer probe was placed in the end of the exhaust tube.

- Tests carried out for a BMW M10B18 and Fiat 1.1 engines fueled with petrol:
 - measurement of the content of nitrogen oxides in exhaust gas (Fig. 10 and Fig. 16),
 - measurement of the hydrocarbon content in the flue gas (Fig. 11 and Fig. 17),
 - measurement of carbon dioxide content in flue gas (Fig. 12 and Fig. 18),

- measurement of the carbon monoxide content in the flue gas (Fig. 13 and Fig. 19),
- measuring the oxygen content in the exhaust gas (Fig. 14 and Fig. 20),
- measurement of the excess air ratio in the exhaust gas (Fig. 15 and Fig. 21).

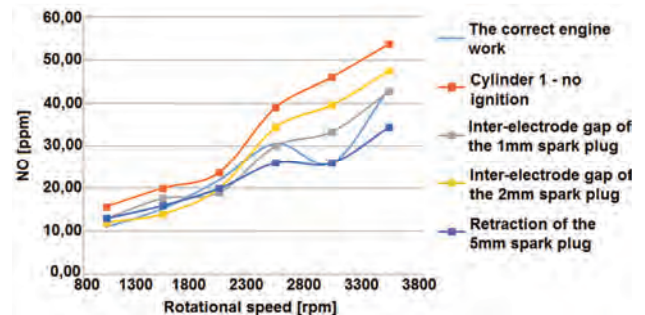


Fig. 10. The content of nitrogen oxides in the exhaust gas of the BMW M10B18 engine

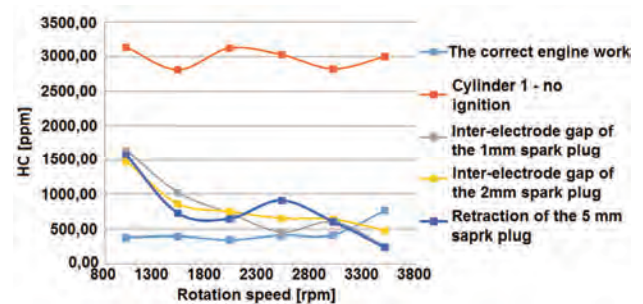


Fig. 11. The hydrocarbon content in the exhaust gas of the BMW M10B18 engine

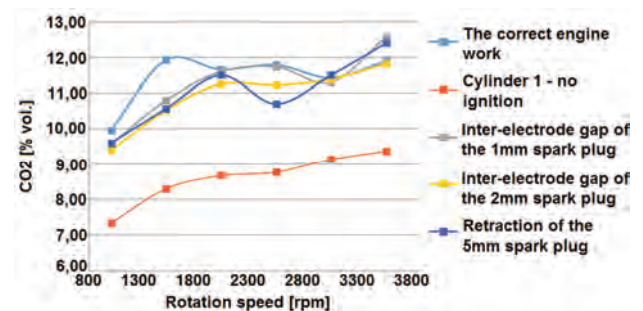


Fig. 12. The carbon dioxide content in the BMW M10B18 engine exhaust gas

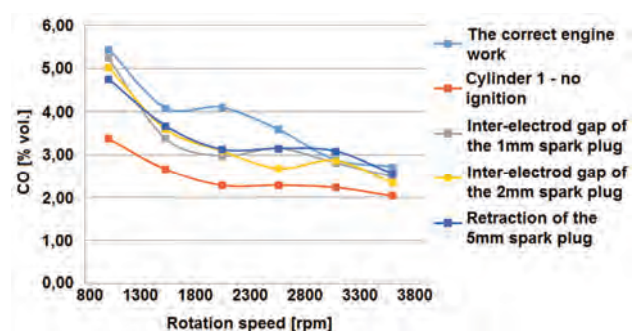


Fig. 13. The content of carbon monoxide in the exhaust gas of the BMW M10B18 engine

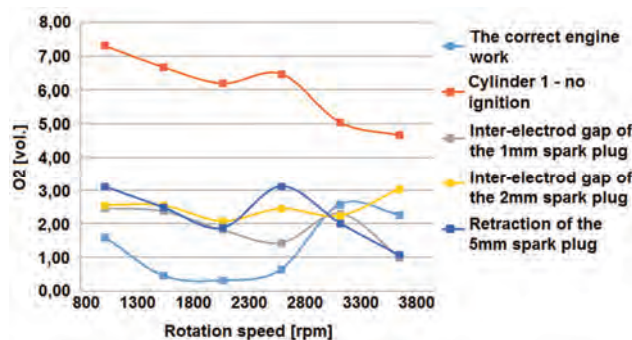


Fig. 14. The oxygen content in the exhaust gas of the BMW M10B18 engine

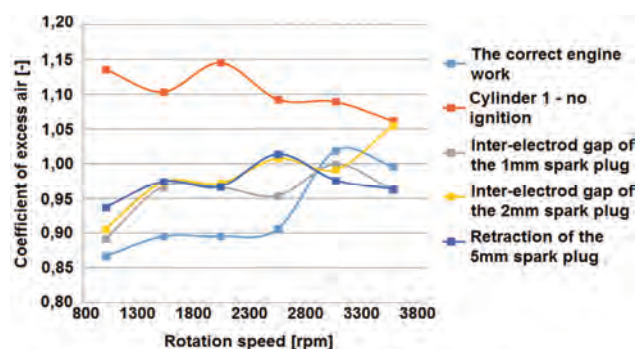


Fig. 15. The value of the BMW M10B18 engine's excess air coefficient

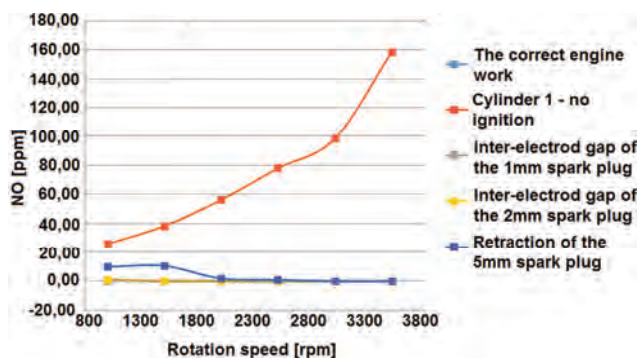


Fig. 16. The content of nitrogen oxides in the exhaust gas of the Fiat 1.1 engine

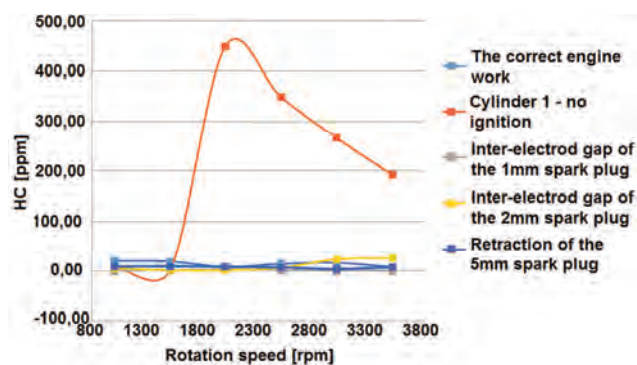


Fig. 17. The hydrocarbon content in the exhaust gas of the Fiat 1.1 engine

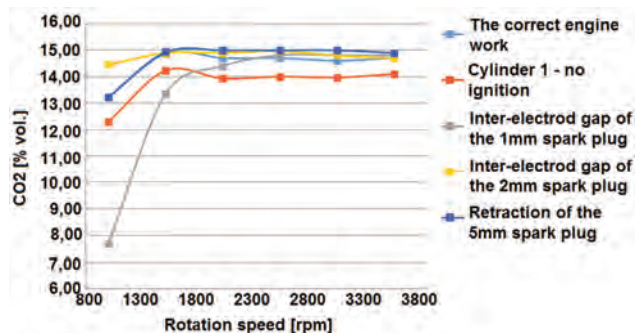


Fig. 18. The carbon dioxide content in the Fiat 1.1 engine exhaust gas

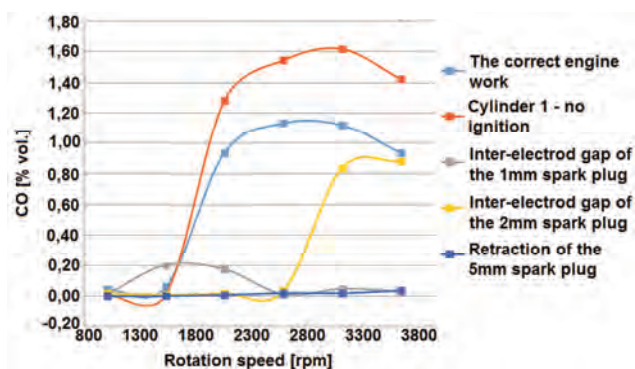


Fig. 19. The content of carbon monoxide in the exhaust gas of the Fiat 1.1 engine

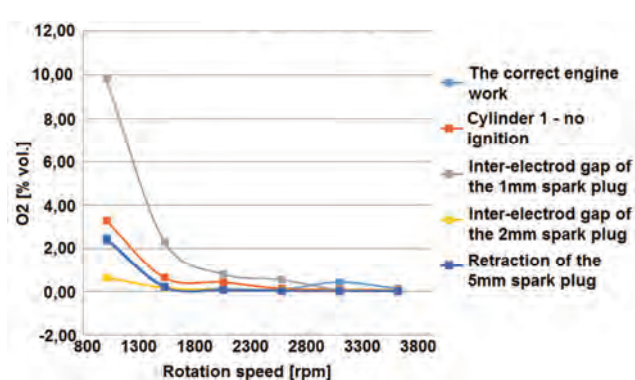


Fig. 20. The oxygen content in the exhaust gas of the Fiat 1.1 engine

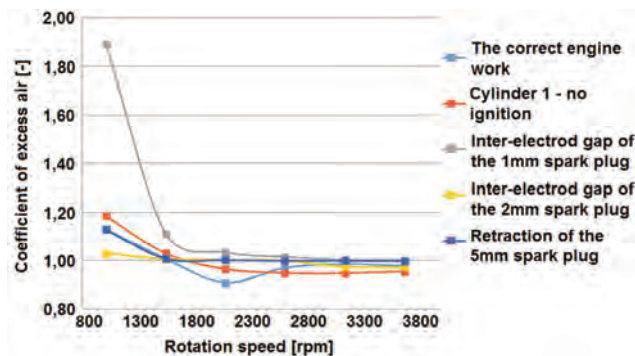


Fig. 21. The value of the Fiat 1.1 engine's excess air coefficient

4.1. Analysis of test results based on surface charts

The mission graphs prepared in the rotational speed co-ordinate system make it easier to interpret the work of the engine and its components. The simulation of basic failures results in engine control changes through the correction of the fuel and air mixture, modification of the ignition advance angle and regulation of the exhaust gas recirculation valve. The settings forced in this way affect the combustion process and the amount of toxic substances emitted. Selected fault conditions occur only for specific engine operation parameters and individual atmospheric conditions.

Analysis of the results obtained allows to observe changes that could have occurred in the engine control in a given speed range for the selected damage.

On the basis of the tests performed, surface charts were developed to facilitate the analysis of results and to determine the dependence between individual engine parameters affecting the emissions of toxic substances. The surfaces created show the dependence of emissions of nitrogen oxides, hydrocarbons, carbon monoxide, carbon dioxide on the rotational speed and the effect of simulated ignition system damages on the toxicity of exhaust gases. The surfaces obtained allow to interpret the results of investigations of changes occurring during the combustion process, which are affected by the engine's operating parameters and the ignition systems used and their associated damage.

The surface charts for the emission of toxic substances for the BMW M10B18 and Fiat 1.1 petrol-fueled engines are shown in Figs 22–25.

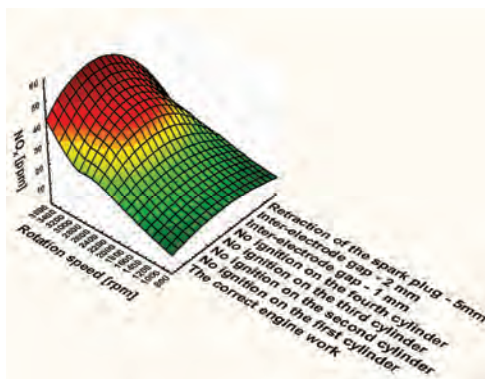


Fig. 22. The content of nitrogen oxides emitted by the BMW M10B18 petrol-powered engine

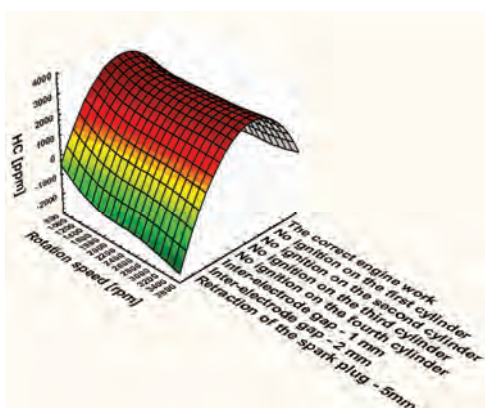


Fig. 23. The content of hydrocarbons emitted by the BMW M10B18 petrol-powered engine

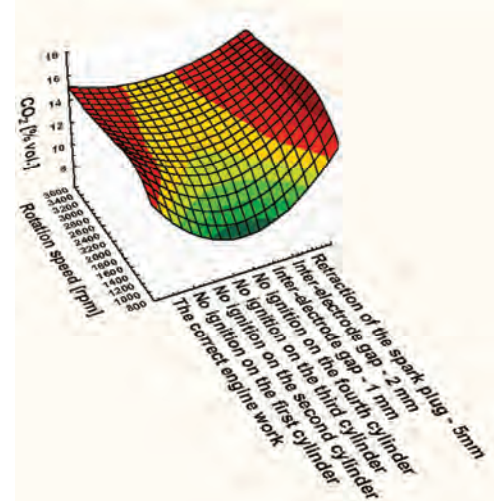


Fig. 24. The carbon dioxide content of the Fiat 1.1 petrol-powered engine

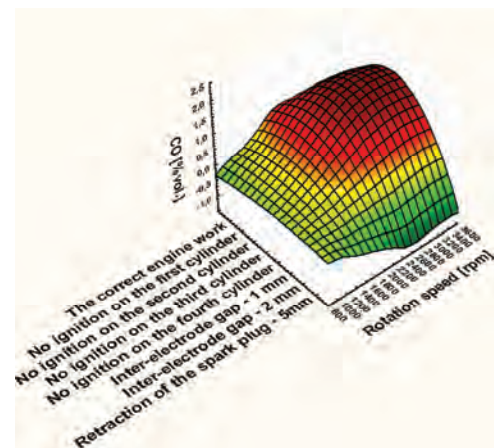


Fig. 25. The carbon monoxide content of the Fiat 1.1 petrol-powered engine

The emission of nitrogen oxides during the correct operation of the BMW engine increases with the increase of the rotational speed. During operation of the engine, 12 ppm nitrogen oxides are emitted at idle speed. For higher speeds from 3500–3600 rpm the emission rises to 35 ppm. Higher values of nitric oxide are achieved when working with an increased inter-electrode gap or offset spark plug. The greatest increase in the emission of nitrogen oxides is visible at the moment of switching off one of the spark plugs. It is twice as high in terms of correct engine operation.

The hydrocarbon emission during the correct engine operation for the whole range of rotational speeds is maintained at 400–500 ppm. Changing the inter-electrode gap or offsetting the spark plug causes emission of hydrocarbons emitted to the value of 1000–1500 ppm. After disconnecting the spark plug, the hydrocarbons increase by a factor of 3000–3500 ppm.

The carbon dioxide released remains at the level of 8% of the exhaust volume during idling. The concentration increases until the rotational speed reaches 2,800 rpm and then remains constant at 12% of the exhaust volume. Increasing the inter-electrode gap or removing the spark plug causes a sharp decrease to 9% of the volume. Excluding

only one of the spark plugs, the hydrocarbon concentration is maintained at the level of 6–7% of the gas volume.

The content of carbon monoxide in the exhaust gases gradually decreases as a function of the rotational speed and reaches its maximum value during idling. A smaller amount of carbon monoxide emitted is obtained by increasing the inter-electrode gap or removing the spark plug. Disconnecting one of the spark plugs causes a decrease in CO to 2–3% of the volume of the exhaust during idling and 1–2% for higher rotational speeds.

5. Summary

The mathematical model of the ignition system developed in the LabView software describes the change of the voltage value on the secondary winding of the ignition coil. It allows to analyse individual sections: sparking, transient and a short circuit of the reference high voltage waveform in relation to the voltage waveform of the simulated ignition system fault.

The tests carried out on the internal combustion engine with a spark ignition proved to what extent the toxicity of exhaust gases changes during simulation of typical ignition system failures. The most unfavourable effects of the engine's impact on the environment occurred during ignition losses, where in one of the cylinders there was no combustion. As a result, the hydrocarbon emission increased several times. By simulating a change in the inter-electrode gap or the removal of the electrode of the spark plug, large changes in the emission of nitrogen oxides and hydrocarbons were observed.

Summarising the analysis of the composition of toxic substances, the impact of damages on the operation of the engine fuelled with gasoline and gas in the content of toxic components was found. Using the EOBD on-board diagnostics and recorded runs of toxic components, a diagnostic database can be developed containing error codes and approximate emission rates.

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Analysis of the impact of wind on fuel consumption and emissions of harmful exhaust gas compounds on the selected flight route

The article discusses the issue of the impact of wind force and direction on fuel consumption and the emission of harmful exhaust gases on the selected flight route. The focus was on percentage changes in fuel consumption and emissions of individual harmful exhaust gas compounds depending on the wind speed and the direction from which it interacts with the aircraft. The analysis was carried out for three different flight levels, in order to compare changes in fuel consumption and emissions also in terms of flight altitude, however the following article focuses only on one level – FL240.

Key words: LTO cycle, impact of wind, emission, fuel consumption

1. Introduction

One of the most modern and the most dynamically developing branches of transport is air transport. Popularity in the selection of these is increasing along with the demand for the transport of goods and people, which is becoming bigger each year. An increase in the wealth of societies, a greater interest in travel and a short flight time as well as a high safety index cause an increase in the number of air connections around the world. According to Airbus forecasts, air traffic doubles every 15 years [2]. The number of flights increased by 80% between 1990 and 2014, and it is predicted that by 2035 it will increase by 45% [4]. The growing number of air connections has a huge impact on the environment and climate change. Aircrafts, in comparison to the other means of transport, emit very large amounts of harmful compounds contained in the exhaust gases, and also generate a high degree of noise, which is particularly undesirable in near-center areas. The following is shown forecasted emission levels of harmful compounds and the forecasted number of flights by 2035 (Fig. 1). Over the past few years, more and more have been heard about the increasing share of CO₂ in the atmosphere and about growing air pollution, which may have disastrous consequences in the future. For example, the total emission of nitrogen oxides in the exhaust gas of aircrafts engines doubled between 1990 and 2014 and is expected to increase by 43% by 2035 [4].

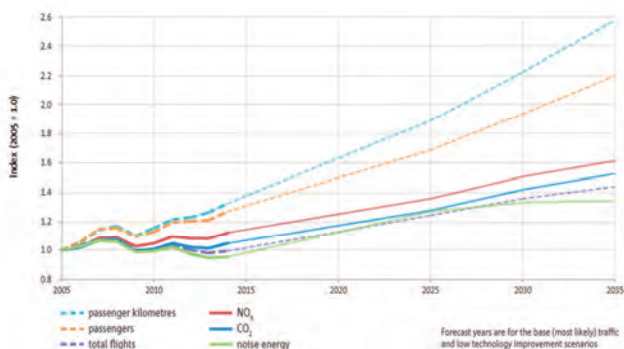


Fig. 1. Noise and emissions forecast [9]

Emission of carbon dioxide between 1990 and 2005 increased by 77%, and between 2005 and 2014 by only 5%. On the other hand, the increase in CO₂ emissions by 44% is estimated between 2005 and 2035 [4]. Considering the aviation sector, which causes a huge amounts of toxic compounds and carbon dioxide during air operations, it is worth analyzing what factors affect the emission of harmful compounds and how it will change depending on the flight parameters. It is known that the longer is the flight time, the more airborne substances are emitted into the atmosphere. It is therefore worth checking how the strength and direction of the wind will affect the speed of the aircraft's flight, shortening or lengthening the flight time, and thus also the emission of harmful exhaust compounds and fuel consumption. For this reason, the following article analyzes different variants of wind direction and force that affects the flight of an aircraft.

The analysis was carried out for three selected flight levels, in order to compare emissions of harmful exhaust compounds and fuel consumption also depending on the flight level, however the following article focuses only on one altitude. The analyzed flight is a flight at FL240 on the Rzeszów–Szczecin route with Dassault Falcon 100 aircraft, equipped with two Garrett TFE731-2-2B engines.

2. Legal regulations concerning the emission of harmful exhaust gases

Along with the growth of air traffic, more and more standards concerning exhaust emissions are imposed, however, tests and measurements of the emissivity of engines focus mainly on operations around the airfield, which may have the greatest impact on people and the environment. The main harmful compounds emitted by turbine aircraft engines are carbon dioxide (CO₂), carbon monoxides (CO), hydrocarbons (HC), nitrogen oxides (NO_x) and particulate matter (PM) [9, 10]. The size of the emission depends on the type of fuel used, the type of engine, the type of aircraft and its mass, as well as on the flight conditions such as altitude, speed and level of thrust [7].

Operations performed by the aircraft can be divided into two groups [7]:

a) operations carried out in the area of the aerodrome: take-off, climb, landing and taxi, which are carried out up to 3000 ft,

b) flight of the aircraft on the route: all operations performed by the aircraft above 3000 ft.

It is worth noting that the largest amount of toxic compounds is emitted in the upper atmosphere, at altitudes from 8 to 12 km above sea level, while only 5 to 10% of world fuel consumption takes place at altitudes not exceeding 1 km above sea level [9]. One of the tests of emission of harmful exhaust gases emitted by civil aircraft is the LTO (Landing and Take off) test. It focuses on operations performed by the aircraft up to the altitude of 3000 ft, and includes such operations as take-off, climb, landing and taxi (Fig. 2).

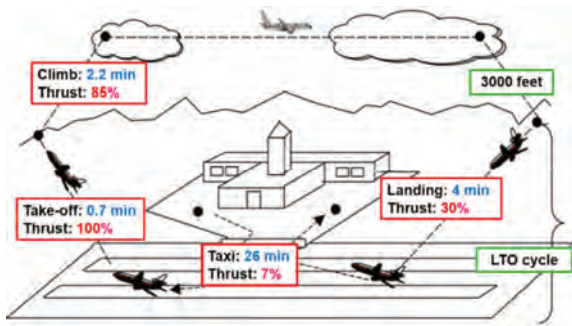


Fig. 2. Flight phases in the LTO cycle [11]

This test includes turbine engines which thrust is more than 27.6 kN. The duration of each test phase corresponds to the generalized time of a particular aircraft operation in real conditions for which a typical engine power setting has also been determined (Table 1). The duration of the whole test is about 30 min [5]. For the proper conduct of the LTO test it is also necessary to measure the average fuel flow rate given in kg/s, emission indexes expressed in g/kg of used fuel for each harmful compound emitted by the engine and information on the measured smoke [13].

Table 1. Engine operating range during the LTO test [13]

LTO work scope	Thrust setting	Phase duration [min]
Take-off	100% Foo	0.7
Climb	85% Foo	2.2
Approach	30% Foo	4.0
Taxi	7% Foo	26.0

The result obtained in the LTO test for a single cycle is the mass of the harmful compound per mass of fuel burnt, which is calculated according to the formula [5]:

$$EPC_{pol,mode} = (TIM/60) \cdot (FFR/1000) \cdot EF \cdot NE \quad (1)$$

where: $EPC_{pol,mode}$ – Emissions per cycle for a particular pollutant during a particular mode [lb/cycle], TIM – Time in Mode [min/cycle], FFR – Fuel Flow Rate [lb/hr], EF – Emission Factor [lb/1000 lb], NE – Number of engines on the aircraft.

To calculate the total emission of a harmful compound it is necessary to sum up the emission value for all phases [5]:

$$EPC_{total} = EPC_{pol,app} + EPC_{pol,taxi} + EPC_{pol,takeoff} + EPC_{pol,climb} \quad (2)$$

In order to calculate the total emissions of harmful compounds emitted during the entire flight of an aircraft, it is also necessary to calculate emissions in the cruising phase. Unfortunately, there are no standards specifying the emission limit values for individual toxic exhaust compounds, i.e. carbon monoxides, hydrocarbons and nitrogen oxides. Most studies on harmful emissions focus only on carbon dioxide emissions, which are almost linearly dependent on fuel consumption [10]. In order to accurately calculate the emission of harmful exhaust gas compounds in the inlet phase, it is necessary to determine emission factors for individual harmful exhaust gas compounds for the pass phase, including flight altitude as well as ambient temperature and pressure in the calculations. Data necessary to calculate emission factors these are emission factors for the LTO cycle in the throughput phase. This solution will allow to obtain coefficients that correspond to a given engine power and external conditions of the flight [8].

The following formulas will be used to determine the emission of harmful compounds:

$$E_{CO} = EI_{CO} \cdot 10^{-3} \cdot K \cdot SFC \cdot t \cdot l \quad (3)$$

$$E_{NO_x} = EI_{NO_x} \cdot 10^{-3} \cdot K \cdot SFC \cdot t \cdot l \quad (4)$$

$$E_{HC} = EI_{HC} \cdot 10^{-3} \cdot K \cdot SFC \cdot t \cdot l \quad (5)$$

where: E_{CO} / E_{NO_x} / E_{HC} – emission of individual substances [kg], EI_{CO} / EI_{NO_x} / EI_{HC} – emission factors for individual substances, depending on the type of engine and the scope of their work [g/kg], K – engine thrust [N], SFC – unit fuel consumption [kg/(N·h)], t – engine run time at a given thrust value [h], l – number of engines on the aircraft.

In order to obtain emission factors of EI harmful compounds, it is necessary to depend them on the temperature, pressure and air humidity at a given cruising altitude:

$$EI_{CO} = EI_{COLTO} \cdot \frac{\theta^{3.3}}{\delta^{1.02}} \quad (6)$$

$$EI_{HC} = EI_{HCLTO} \cdot \frac{\theta^{3.3}}{\delta^{1.02}} \quad (7)$$

$$EI_{NO_x} = EI_{NO_xLTO} \cdot \sqrt{\frac{\delta^{1.02}}{\theta^{3.3}}} \cdot e^h \quad (8)$$

where: EI_{CO} , EI_{HC} , EI_{NO_x} – CO, HC and NO_x emission factors at a given altitude [g/kg], EI_{COLTO} , EI_{HCLTO} , EI_{NO_xLTO} – CO, HC and NO_x emission factors measured for LTO parameters [g/kg].

To calculate the temperature, pressure and humidity coefficients, the following formulas should be used:

θ – temperature change coefficient [–]:

$$\theta = \frac{T_c}{288.15 \text{ K}} \quad (9)$$

δ – pressure change coefficient [–]:

$$\delta = \frac{P_c}{101325 \text{ Pa}} \quad (10)$$

e – Euler number ($e = 2.72$)

h – the air humidity factor depends on the altitude [–]:

$$h = -19 \cdot (\omega - 0.00634) \quad (11)$$

ω – specific humidity,

that: $\omega = 10^{-3} \cdot e^{-0.0001426 \cdot (H-12900)}$,

where H is the cruising altitude given in feet.

In turn, to determine the parameters of the environment at a given speed and flight altitude necessary to calculate the temperature and pressure change coefficients, use the dependence [8]:

$$T_c = T \cdot (1 + 0.2 \cdot Ma^2) \quad (12)$$

$$p_c = p \cdot (1 + 0.2 \cdot Ma^2)^{3.5} \quad (13)$$

where: T_c – total temperature [K], p_c – total pressure [Pa], T – ambient temperature [K] at a given flight altitude (according to the standard atmosphere), p – ambient pressure [Pa] at a given flight altitude (according to the standard atmosphere), Ma – airplane speed [–].

To calculate T and p , use the parameters of the reference atmosphere and the dependence between them. To determine the temperature at a given cruising altitude, a vertical temperature gradient of up to 11,000 m must be used, i.e. a temperature drop of 6.5 Kelvin for every kilometer of altitude, from the sea level, i.e. equal to 288.15 K. From 11,000 m up to 20,000 m, the temperature is constant and amounts to -56.5° C. In the case of determining the pressure at a given flight altitude, use the formula [12]:

$$p = p_0 \cdot \left(1 - \frac{H_1}{44300}\right)^{5.256} \quad (14)$$

where: p_0 – pressure at sea level equal to 1013.25 hPa, H_1 – flight altitude expressed in meters.

3. Research methodology

- In order to analyze the impact of wind on fuel consumption and emissions of harmful exhaust compounds, it was needed to make assumptions necessary to perform calculations. A few more important assumptions are presented below:
- For the purpose of determining the length of the route that the aircraft will defeat in relation to the ground during the climb to a specific cruising altitude, a climb angle of 10° was assumed for the entire aircraft's climb phase. This angle was established on the basis of a list of the construction angles of selected Boeing aircraft [1].
- In order to calculate the emission of harmful exhaust gas compounds, i.e. hydrocarbons, carbon monoxides and nitrogen oxides, the same relationships as for the calculation of emission of harmful exhaust gas in the LTO test were used. Due to the lack of information about the emission factors of individual harmful exhaust gas compounds, the percentage setting of the engine thrust and the average fuel flow rate at the aircraft height above 3000 ft (about 1000 m) data used in the ICAO Database [6] were used for LTO tests the engine being analyzed. This means that for an aircraft altitude of over 3000 ft, the engine thrust setting is 85% of the nominal thrust. However, the average fuel flow rate is 0.173 kg/s and used to calculate the fuel consumption during the climb phase.
- The flight time from the starting point to the ending point shall not take into account the effect of wind on the change of direction of flight and the course of the aircraft, which results in a longer journey and flight time. For this reason, in the analysis of the impact of wind force and direction on the speed of the aircraft flight, the wind component perpendicular to the direction of flight of the aircraft is not taken into account. It is assumed that the wind speed and assumed wind direction do not change during the entire flight phase.
- For the problem analysis, an engine thrust of 50% of the nominal thrust was assumed for the in-flight phase.
- In order to calculate the emission of harmful exhaust gases during the flight phase, it is necessary to know the emission factors of individual harmful compounds, which coefficients have not been determined by engine manufacturers or are not available to aircraft users. Therefore, the emission ratios of harmful compounds for individual phases of the LTO test and on their basis a graph was drawn up taking into account the degree of setting the engine thrust were used. On the basis of the trend line of a given graph, the estimated value of the emission factor of a specific harmful compound for the engine set-up equal to 50% of the nominal thrust was calculated.
- In order to determine the distance that the aircraft will travel to the ground during descent from a specific cruising altitude, a descent angle of 3° is assumed for a landing approach operation starting at 500 ft (approximately 152 m). For the landing phase between the cruising altitude and the landing altitude of the landing approach, a descent angle of 10° was assumed.
- To calculate emissions of harmful exhaust compounds, i.e. hydrocarbons, carbon monoxides and nitrogen oxides during the climb and landing phase, the same relation as in the case of calculating emissions of harmful exhaust gas in the LTO test has been applied. Due to the lack of information on the emission factors of individual harmful exhaust gas compounds, the percentage setting of the engine thrust and the average fuel flow rate at elevation and descent of the aircraft above 3000 ft (about 1000 m), the data contained in the ICAO Data Base [6] were used for LTO tests of the analyzed engine. This means that for an aircraft altitude of over 3000 ft, the engine thrust setting is 85% of the nominal thrust. However, the average fuel flow rate is 0.173 kg/s and used to calculate the fuel consumption during the climb phase. However, for the landing phase starting at an altitude greater than 3000 ft, the engine thrust setting is 30% of the nominal thrust. The average fuel flow rate is 0.067 kg/s.
- For the purposes of the analyzes, a weak wind of 10 kt, a moderate wind of 25 kt and a strong wind of 40 kt was assumed.
- In order to make it easier to compare results and handle the cases of wind force and direction considered, the following designations were introduced:
 - Wind direction:
 - P** – the wind blowing in the forehead of the aircraft,
 - T** – wind blowing in the tail of the aircraft.
 - Angle of wind (towards the direction of flight of the aircraft):
 - 1** – 0 degrees,
 - 2** – 30 degrees,

3 – 45 degrees,

4 – 60 degrees.

The parameters of the selected aircraft and its engine are shown in Table 2 and 3.

Table 2. Dassault Falcon 100 aircraft performance (own elaboration based on [3])

Range	2446 km
MTOW	8500 kg
BEM	4880 kg
Max cruising speed	907 km/h
Max cruising altitude	13716 m

Table 3. Parameters of the Garrett TFE731-2 engine (own elaboration based on [12])

Max thrust	15.6 kN
Fuel consumption	396.89 kg/h
SFC	0.05 kg/Nh
The degree of compression	13:1

The specific fuel consumption (SFC) was calculated basing on the fuel consumption and the half value of the maximum thrust (the cruise flight use only the half power of engines). Also the source [1] shows the SFC as 0.5 lb/lbf/h and the source [6] shows the SFC as 0.083 kg/Nh, so after recounting this valuables on the demand units and comparing it with the calculated one from the definition of SFC, the value of SFC was adopted as 0.05 kg/Nh.

4. Analysis of flight in windless conditions at FL240

Calculations of fuel consumption and emission of harmful exhaust gases for windless conditions on the entire route from Rzeszow to Szczecin were made, the results of which provide a certain basis for references for further calculations and analyzes. The analysis includes the climb phase, the cruise phase, the landing phase and all ground operations that are included in the LTO test. The results are presented in the form of diagram in Fig. 3.

According to the results, fuel consumption in the cruise phase constitutes about 70% of the total fuel consumed during the flight. During the climb phase, fuel consumption is about 12.5% of the total fuel consumed, and during the approach phase it is about 5%. Fuel consumption during the LTO test is constant regardless of the phases and amounts to approximately 11% of the total fuel consumed during the flight. It can be seen that the least fuel is consumed during the approach phase of the aircraft. In the case of analysis of emissions of individual toxic compounds in specific flight phases for different altitudes of aircraft (Fig. 3b), it can be noticed that the largest part of carbon monoxides (93%) and nitrogen oxides (79%) is emitted during the flight phase because it lasts the longest. The second longest phase is operations included in the LTO test, in which the emission of carbon monoxides accounts for about 15%, while the emission of hydrocarbons in this phase is the largest and constitutes about 28%.

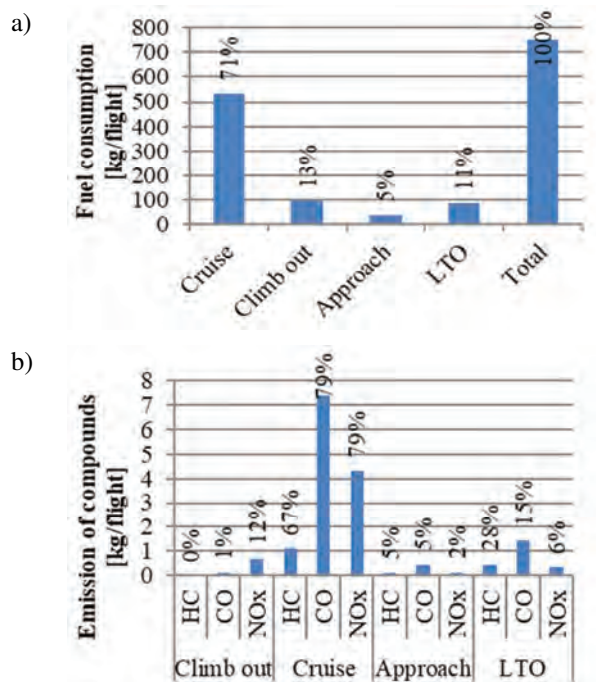


Fig. 3. Results in individual stages of the flight, a) fuel consumption, b) emission of harmful exhaust compounds

5. Analysis of the cruise phase at FL240

5.1. Analysis for low wind speed (10 kt)

Calculations of fuel consumption and emissions of harmful exhaust compounds for different variants of wind speed and direction on one flight level were made. The results are illustrated in the graphs and referred to windless conditions at a given flight level.

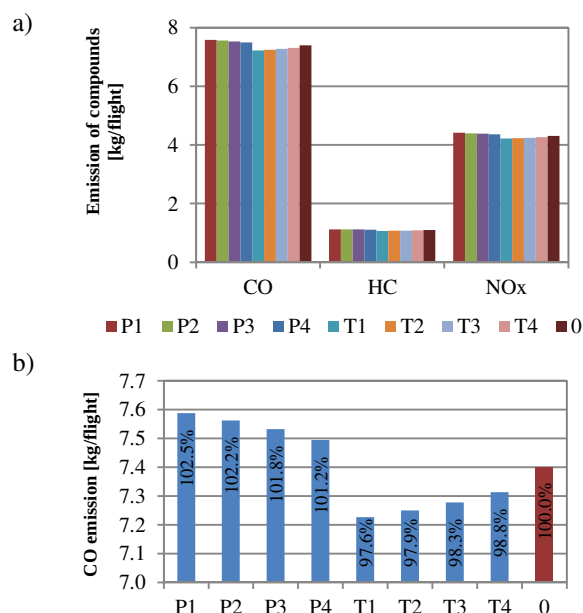


Fig. 4. Emission of harmful exhaust fumes in a given case, a) in general b) emission of carbon monoxides CO

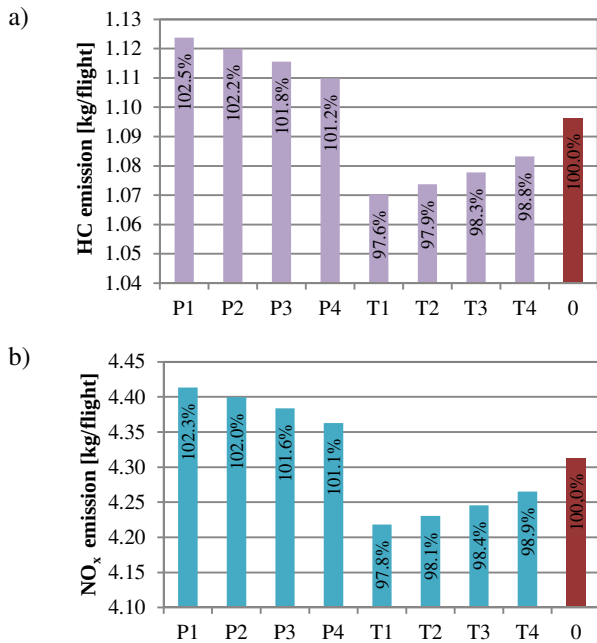


Fig. 5. Emission of harmful exhaust fumes in a given case, a) hydrocarbons HC, b) nitrogen oxides NO_x

The summary chart (Fig. 4a) shows that the differences in the emission of individual harmful compounds between P1-T4 cases are small. The exact percentages can be read from the following charts (Fig. 4b and Fig. 5). The difference between the wind that is the most favorable for the flight (T1) and the least favorable wind (P1) is 4.9% for the emission of carbon monoxides and hydrocarbons in accordance to the emission in still conditions, whereas in the emission of nitrogen oxides the difference is 4.5%. It can also be seen that as the angle of the wind is increased relative to the direction of flight of the aircraft, the emission of toxic compounds slightly decreases with respect to the wind blowing from the front of the aircraft. However, in relation to the wind blowing in the tail of the aircraft, increasing the winding angle causes an increase in the emission of harmful exhaust gases. The most favorable in terms of emissions is therefore the wind blowing from the back of the aircraft, which can reduce the emission of carbon monoxides, hydrocarbons and nitrogen oxides by 2.4%, 2.4% and 2.2% respectively in relation to the flight in windless conditions. In the case of carbon monoxide, the difference between the most favorable wind and windless conditions is about 0.18 kg of carbon monoxide per passage. In the case of hydrocarbons, it is about 0.026 kg per passage, and for nitrogen oxides the difference is about 0.09 kg per flight.

Analyzing the fuel consumption (Fig. 6a) for the FL240, the same results can be observed as in the case of emissions of individual harmful compounds, i.e. compared to the windless conditions for the wind blowing in front of the aircraft, as the winding angle decreases with respect to the flight direction of the aircraft, the fuel consumption increases, reaching a maximum of 2.38% more for P1. In the case of wind blowing from the rear of the aircraft, with increasing wind angle relative to the direction of flight, fuel consumption increases, the smallest for wind blowing in the direction of flight, and is 2.27% less compared to windless

conditions, which gives about 12.14 kg of fuel less. Focusing on the differences in the speed of the aircraft's approaching to the ground (Fig. 6b), it can be also seen about a 2% drop in speed for wind blowing from the front of the aircraft at different angles, and about 2% increase in speed for wind blowing in the tail. In the most favorable flight conditions (T1), this speed increases by 2.3% relative to windless conditions, which gives a ground speed of 816.56 km/h. These speeds directly affect the flight time between Rzeszow and Szczecin (Fig. 6c) and in the case of T1 allow to shorten the flight by 2.3% relative to windless conditions, so by 0.9 minutes. Differences between the most favorable conditions (T1) and the least favorable conditions (P1) at the aircraft speed are 4.6% relative to the windless conditions, which translates into 2.76 min.

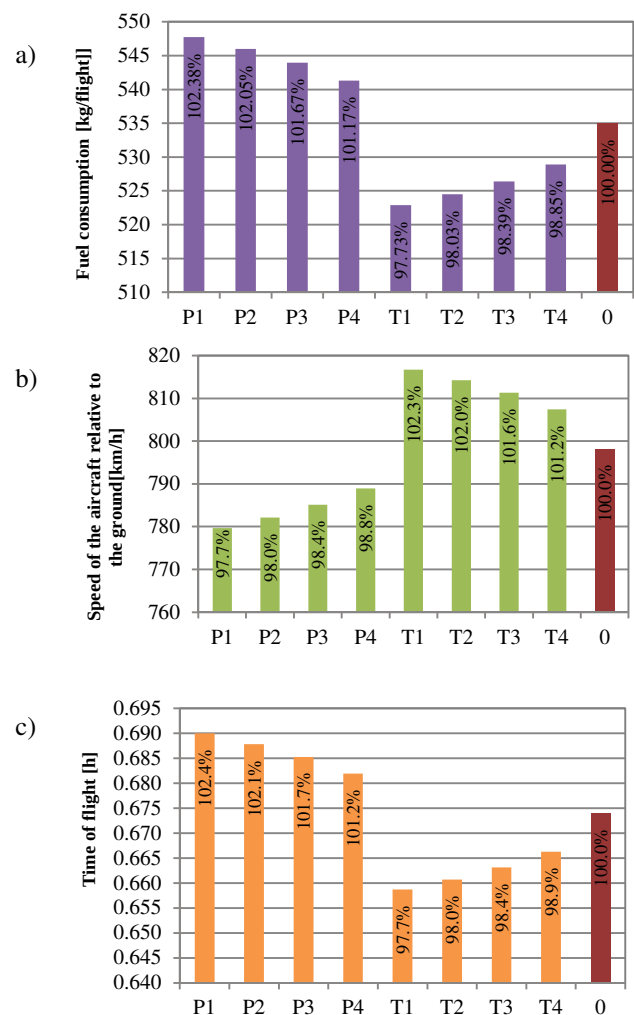


Fig. 6. Analysis of differences in main parameters a) fuel consumption, b) speed of the aircraft relative to the ground, c) flight time

For the analyzed flight level at a wind speed of 10 kt, the plane will fly a route from Rzeszow to Szczecin with the least beneficial wind conditions in time 41.4 min at a speed of 779.68 km/h and will use 547.73 kg of fuel. In the most favorable wind conditions, the aircraft will cover the designated route within 39.54 minutes at a speed of 816.72 km/h and consumes 522.89 kg of fuel.

5.2. Analysis for moderate wind speed (25 kt)

Calculations of fuel consumption and emissions of harmful exhaust compounds for 25 kt wind strength and different direction on one FL240 were made. The results are illustrated in the graphs and referred to windless conditions.

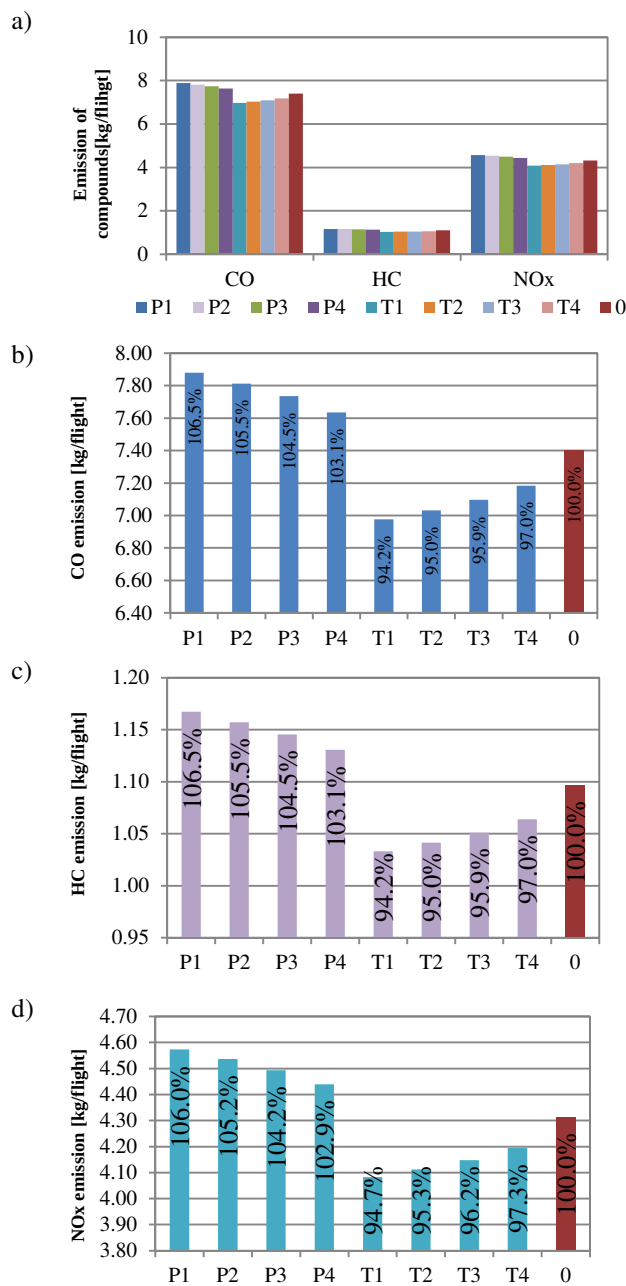


Fig. 7. Emission of harmful exhaust fumes in a given case, a) in general, b) emission of carbon monoxides CO, c) hydrocarbons HC, d) nitrogen oxides NOx

In Figure 7a slight differences between the emission of individual harmful compounds for the variable wind angle were shown. Hydrocarbon emission in each of the analyzed cases is the lowest, and the largest is carbon monoxides. From the following graphs, the exact percentages between the individual variant P1–T4 and the windless conditions can be read.

The difference in the emission of carbon monoxides (Fig. 7b) for the most favorable (T1) and least favorable conditions (P1) is 12.3% in relation to windless conditions. This gives a difference equal to 0.91 kg of carbon monoxides. Thus, in the least favorable conditions, the emission of carbon monoxides will be by 0.48 kg of carbon monoxides more than in the case of windless conditions, while in the case of the most favorable conditions, this emission decreases by 0.43 kg compared to the windless conditions. The hydrocarbon emission results are similar (Fig. 7c), for which the difference between P1 and T1 is also 12.3% relative to windless conditions, which gives about 0.13 kg. In the case of the most favorable conditions, the aircraft will emit 6.5% more hydrocarbons, or 0.07 kg more than in the case of windless conditions, and in the case of the least favorable conditions it will emit 5.8% less, or 0.06 kg of hydrocarbons less. Some other percentages take the emission of nitrogen oxides (Fig. 7d), for which the difference between P1 and T1 is 11.3%, so it is the smallest in comparison to other harmful compounds. In the case of P1, the aircraft will emit 6% more nitrogen oxides than in windless conditions, which gives 0.25 kg. In the case of T1, the difference is 5.3% less than in windless conditions, equivalent to 0.23 kg of carbon monoxides.

Comparing the fuel consumption for the analyzed conditions (Fig. 8a), a 6.2% difference is visible in the least favorable conditions compared to windless conditions.

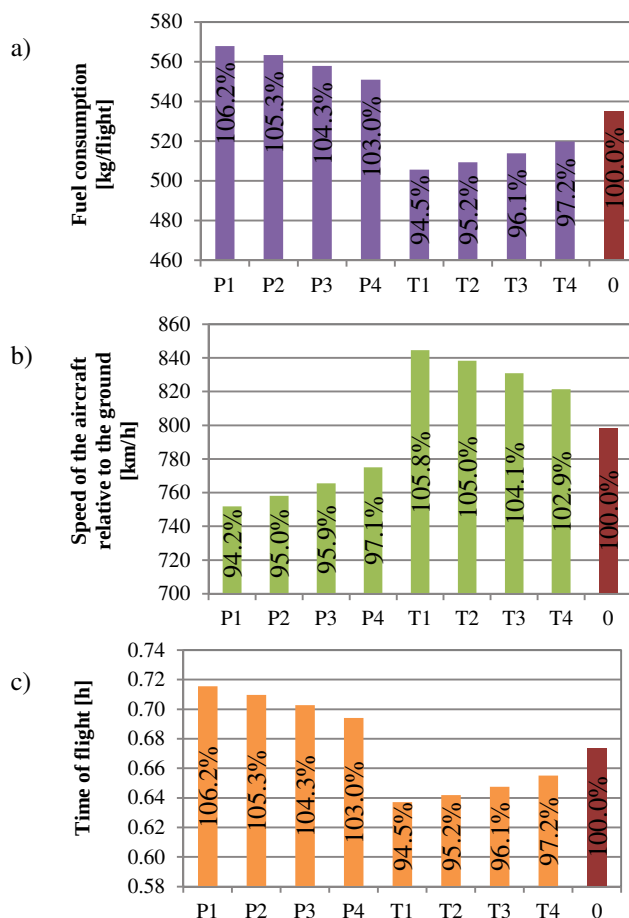


Fig. 8. Analysis of differences in main parameters a) fuel consumption, b) speed of the aircraft relative to the ground, c) flight time

Means, that the aircraft will use 33.17 kg more fuel when the wind of moderate speed is blowing from the front of the aircraft. Under the most favorable conditions, fuel consumption will decrease by 5.5% relative to the windless conditions, which gives 29.4 kg less fuel than the windless conditions in the case of T1.

Comparing the change in the ground speed (Fig. 8b) in the case of a change in the wind direction, the aircraft speed decreases by around 1% with the change of the wind angle from the highest to zero flying from the plane, and the speed increase by about 1% when changing wind angle from the highest to zero for wind blowing in the direction of flight. The speed of the aircraft will decrease by 5.8% in the case of P1 compared to the windless conditions, which gives 46.3 km/h less. However, in T1 conditions, this speed will increase by 5.8%, so the aircraft will move by 46.3 km/h faster, which shortens flight time (Fig. 8c) by 0.037 h, or about 2.22 min. The flight time of the aircraft varies in proportion to the change in the speed of the aircraft's flight relative to the ground.

5.3. Analysis for strong wind speed (40 kt)

When comparing the wind direction change with the highest of the analyzed speeds (40kt) at the FL240 level, it can be noticed that the emission of harmful exhaust compounds (Fig. 9a) is very similar to the wind velocities previously analyzed. In the case of carbon monoxide emissions (Fig. 9b), it is clear that the emission variation for particular wind direction cases is the highest of all wind speeds analyzed.

The difference between P1 and T1 is as much as 19.6% relative to windless conditions. Under the least favorable conditions, the aircraft will emit by 10.7% more carbon monoxides than in still conditions, which gives a value equal to 0.79 kg. However, under the most beneficial conditions, it will emit 8.9% less carbon monoxides than in windless conditions, which gives about 0.66 kg.

Analyzing hydrocarbon emissions (Fig. 9c), one can also notice the greatest variation between individual cases of P1–T4 compared to the previously analyzed wind speeds. In the case of P1, the hydrocarbon emission will increase by 10.7% compared to the windless conditions, so by 0.12 kg. In the case of T2, the hydrocarbon emission will decrease by 8.9% relative to the windless conditions, which gives about 0.1 kg.

Differences in nitrogen oxide emissions (Fig. 9d) are as follows: 10% (0.43 kg) more hydrocarbons for P1 compared to windless conditions, and 8.3% (0.35 kg) of hydrocarbons less in T1 case compared to windless conditions.

Analyzing the fuel consumption (Fig. 10a) the difference between P1 and T1 is as much as 18.7%, which gives 100 kg of fuel. Under the most favorable conditions, fuel consumption is 489.58 kg, i.e. 8.5% less than in windless conditions (535 kg). However, in the least beneficial conditions, fuel consumption is 10.2% more, that is by as much as 54.6 kg more.

Differences in the speed of the aircraft against the ground (Fig. 10b) are also the highest in comparison with the previously analyzed aircraft speeds. In the case of P1, the speed of the aircraft will decrease by 9.3%, while in the

case of T1 this speed will increase by 9.3%, which gives a difference of 74.23 km/h depending on the wind direction.

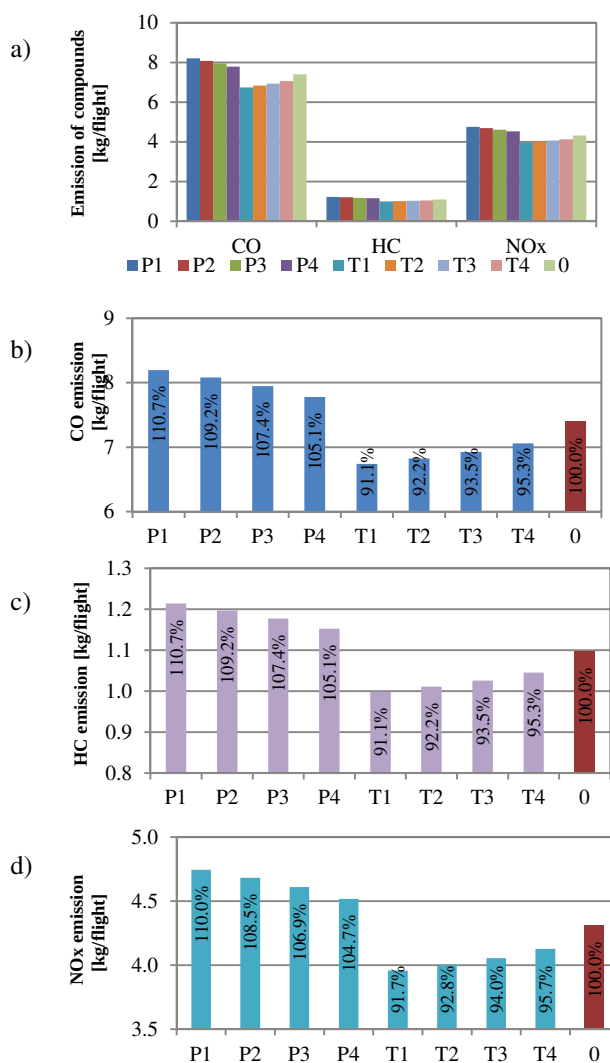


Fig. 9. Emission of harmful exhaust fumes in a given case, a) in general, b) emission of carbon monoxides CO, c) hydrocarbons HC, d) nitrogen oxides NO_x

The flight time of the plane (Fig. 10c) will be 0.617 h or 37.02 minutes in the most favorable conditions, while in the least beneficial cases it will be 0.743 h, or 44.58 minutes. It is also evident that the flight time of an aircraft increases as the angle at which the wind in the wind blows increases (T1–T4), and that with the increase of the wind angle in relation to the direction of flight of the aircraft (P1–P4) the flight time decreases aircraft.

6. Conclusions

In the article influence of wind force and direction on fuel consumption and emission of harmful exhaust gas compounds during the flight were presented. The analysis was carried out for three selected flight levels, in order to compare emissions of harmful exhaust compounds and fuel consumption also depending on the flight level, however the article focuses only on one altitude. The analyzed flight taken place at FL240 on the Rzeszow–Szczecin route with

Dassault Falcon 100 aircraft, equipped with two Garrett TFE731-2-2B engines.

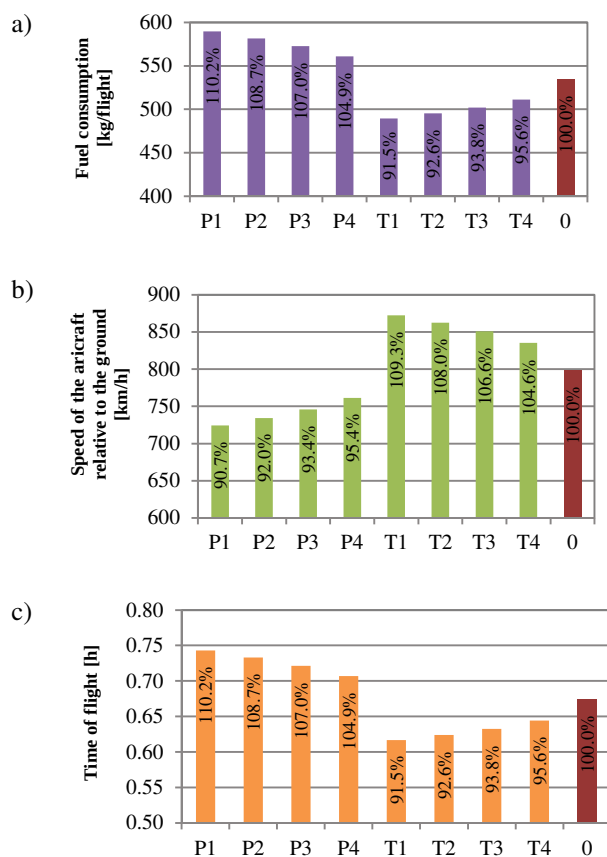


Fig. 10. Analysis of differences in main parameters a) fuel consumption, b) speed of the aircraft relative to the ground, c) flight time

The performed calculations allowed for a percentage comparison of fuel consumption and emissions of individual toxic exhaust compounds depending on the strength and direction of the wind, and in windless conditions. The list of the most favorable calculation results depending on the wind speed compared to windless conditions is presented below (Table 4).

Table 4. Total fuel consumption and emission of harmful compounds for windless conditions and for windy conditions

Wind	Fuel consumption [kg/flight]	CO [kg/flight]	HC [kg/flight]	NO _x [kg/flight]
No wind	749.59	7.95	3.10	5.46
Strong tail	704.15	7.29	3.00	5.10

The comparison shows that at the FL240 flight level, the most favorable fuel consumption and emission of harmful exhaust compounds is achieved by the aircraft during the flight with a wind speed of 40 kt which blows from the tail of the aircraft. The fuel consumption is then reduced by 14% in relation to the flight with the same wind but blowing from the front of the aircraft, and by 6% relative to the windless conditions. Also the emission of harmful gas compounds is the smallest when the wind is the strongest from all analyzed speeds of wind, and when this wind is a rear-wind. In comparison to the windless conditions, emission of CO is 9% lower when the wind is rear and strong, emission of HC is 3% lower and emission of NO_x is about 7% lower. That shows, how big impact on fuel consumption and also on emission of harmful gas compounds has the speed and direction of wind, which affect also on the environment.

Nomenclature

BEM	Basic Empty Mass [kg]
CO	carbon monoxide
CO ₂	carbon dioxide
E _{CO}	carbon monoxide emission [kg]
EF	Emission Factor [lb/1000 lb]
E _{HC}	hydrocarbons emission [kg]
EI _{CO}	emission factor of carbon monoxide [g/kg]
EI _{COLTO}	CO emission factors measured for LTO parameters [g/kg]
EI _{HC}	emission factor of hydrocarbons [g/kg]
EI _{HCLTO}	HC emission factors measured for LTO parameters [g/kg]
EI _{NO_x}	emission factor of nitrogen oxides [g/kg],
EI _{NO_xLTO}	NO _x emission factors measured for LTO parameters [g/kg]
E _{NO_x}	nitrogen oxides emission [kg]
EPC _{pol,app}	emission for a single cycle (EPC) for a particular harmful exhaust gas compound (pol) in a approach phase [lb/cycle]
EPC _{pol,climb}	emission for a single cycle (EPC) for a particular harmful exhaust gas compound (pol) in a climbing phase [lb/cycle]

EPC _{pol,mode}	emission for a single cycle (EPC) for a particular harmful exhaust gas compound (pol) in a specific phase (mode) [lb/cycle]
EPC _{pol,takeoff}	emission for a single cycle (EPC) for a particular harmful exhaust gas compound (pol) in a taking off phase [lb/cycle]
EPC _{pol,taxi}	emission for a single cycle (EPC) for a particular harmful exhaust gas compound (pol) in a taxi phase [lb/cycle]
EPC _{total}	total emission of a harmful compound [lb/cycle]
FRR	Fuel Flow Rate [lb/hr]
NE	Number of engines on the aircraft
h	the air humidity factor depends on the height
H	cruising altitude [ft]
H ₁	flight altitude [m]
HC	hydrocarbons
ICAO	International Civil Aviation Organization
K	engine thrust [N]
l	number of engines [-]
LTO	Landing and take-off cycle, model take off and landing cycle used to determine emis-

	sions of noxious exhaust components of an aircraft engine	T	ambient temperature at a given flight altitude [K]
Ma	Speed of aircraft [–]	t	engine run time at a given thrust value [h]
MTOW	Maximum Take-off Weight [kg]	T_c	total temperature [K]
NO_x	nitrogen oxides	TIM	Time in Mode [min/cycle]
p	ambient pressure at a given flight altitude [Pa]	δ	pressure change coefficient [–]
p_c	total pressure [Pa]	θ	temperature change coefficient [–]
PM	particulate matter	ω	specific humidity [–]
p_o	pressure at a level equal to the sea level [Pa]		
SFC	Specific Fuel Consumption		

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Comparative analysis of pollutants emission by classical and distributed propulsions applied on the AOS motor glider

Comparative analysis of harmful compounds emission of classical and distributed propulsions applied on the AOS motor glider, taking into account the perspective of the development of hybrid propulsions. A novel path is indicated by so-called distributed aircraft propulsion. The advantages and disadvantages of this type of solutions are presented, as well as the conceptual design of the distributed propulsion for the AOS 71 motor glider. In the paper there were compared the emissions of harmful compounds generated by a hybrid power unit developed for the airframe of AOS 71 motor glider – traditional propulsion, so-called focused (one-propeller) and dispersed propulsion (multi-propelled). Functional diagrams of both types of propulsions solutions are presented. Construction and aerodynamic constraints of both propulsions are discussed and comparative analysis is made. In the traditional version of the propulsion (so-called focused propulsion), the propeller is driven by an Emrax 228 electric engine with effective parameters: $N = 55 \text{ kW}$, $M = 120 \text{ Nm}$. The power source is a battery set with a capacity of 16 Ah and a range extender powered by a LCR 407ti rotating piston engine with maximum power of 28 kW. In the variant of the distributed propulsion. Ten electric engines of AXI 8120 type were used to drive small propellers arranged along the wingspan. The power source in this variant is analogous to the variant with the Emrax electric engine. For the adopted variant of the flight mission of the motor glider, a flight trajectory model was developed, which was used to determine the load of the power unit. In laboratory conditions, emission tests of both propulsions were conducted. The results are summarized in charts and discussed in the conclusions.

Key words: distributed propulsion, hybrid, motor glider, rotary engine, Wankel engine, hybrid propulsion, emission, exhaust gases

Introduction

The electric and hybrid propulsion is becoming a more and more interesting alternative to conventional propulsion systems used in aircraft. In the case of new types of constructions there appear the problems with increased mass of the power unit or limited range due to low battery capacity [1, 19]. To improve the parameters of electric or hybrid propulsion, research on distributed propulsion is being conducted [1, 19]. Figure 1 presents the concept of an experimental LeapTech aircraft (a consortium of LeapTech and NASA) equipped with an electric distributed propulsion. According to calculations made by the aircraft's designers, the lift coefficient increased by 3 times due to the appropriate aerodynamic design of the propulsion – acceleration of the stream over the wing.

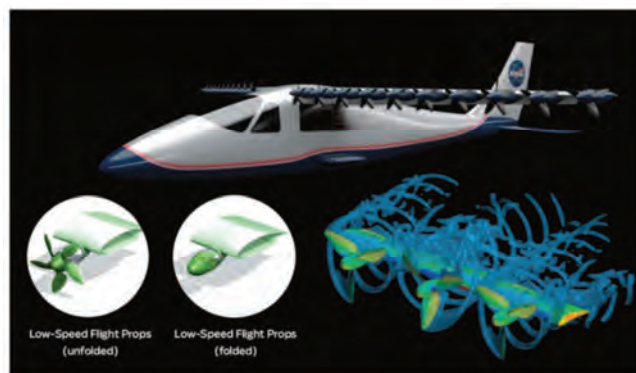


Fig. 1. The LeapTech aircraft [1]

Figure 2 shows the type of the UAV (Unmanned Aerial Vehicle) of the NASA construction, enabling vertical start.



Fig. 2. The UAV of NASA construction[19]

The distributed propulsion is based on the use of a series of low power engines placed along the wingspan or in the appropriate places of the hull. The total thrust generated by such a power unit is sufficient for the safe operation of the aircraft. In addition, the method of integrating the engines with the airframe can affect the aerodynamic performance of the aircraft. It is also possible to reduce or eliminate control surfaces by properly differentiating (vectoring) the engines thrust. The conducted research brings interesting results (improvement of aerodynamic parameters – increase of the lift coefficient, etc. [1, 14]).

The paper describes a comparative analysis in terms of both pollutant emissions and performance for the hybrid power unit of the AOS-71 motor glider of classical and distributed propulsion.

1. Research object

As the basis for the analysis the airframe of the AOS-71 electric glider (Fig. 3) was assumed. The aircraft was built in cooperation between the Department of Aircraft and Aircraft Engines at the Rzeszow University of Technology

and the Faculty of Power and Aeronautical Engineering at the Warsaw University of Technology.



Fig. 3. The AOS 71 motor glider

In Table 1 the basic data of the airframe are presented, and in Fig. 4 the values of power required for the flight in relation to the flight speed.

Table 1. Basic data of the AOS 71 motor glider) [11, 12]

Wing area	$S [m^2]$	15.8
Wing span	$R [m]$	16.4
Aspect ratio	Λ	17
Maximum take-off mass	$M_{max} [kg]$	660
Minimum motor glider mass	$M_{min} [kg]$	500

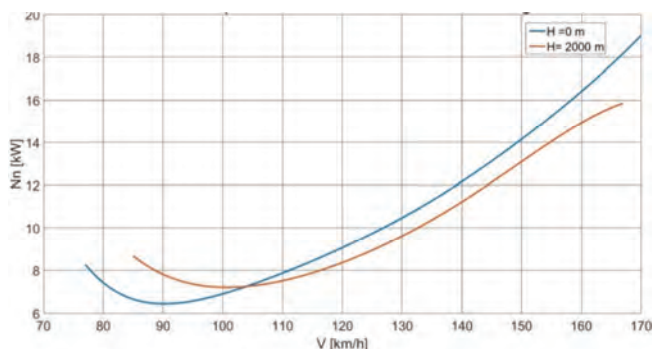


Fig. 4. Power required for flight in the function of flight speed – for the AOS 71 motor glider; $M_{max} = 660$ kg [11, 12]

The power source for the analysed propulsions is a hybrid system in which a small battery set cooperates with an electric generator driven by a rotary piston (Wankel AG 407TGi). Table 2 shows the technical data of the engine and battery assembly.

Table 2. Technical data of power unit [11, 12, 16]

Engine		407TGi
Maximum engine power	$N_{max} [kW]$	31.5
Maximum torque	$M_{max} [Nm]$	51
Engine mass	$m_s [kg]$	20
Specific fuel consumption	$SFC_{min} [kg/kWh]$	0.3
Battery	–	Li-Pol
Capacity	$C [Ah]$	16
Voltage	$U_{bat} [V]$	355
Battery mass	$m_{bat} [kg]$	50

For the internal combustion engine, the rotational characteristics was determined in order to adjust properly the range of its work to the generator's work (Emrax 188). In addition, the emission of pollutants in the exhausts of the 407TGi engine was determined in the function of its rotational speed with used Horiba analyzer. Measurements were made on the research stand in the Department of Aircraft and Aircraft Engines. Figures 5–9 show the test stand, the rotational characteristics of the internal combustion engine and the NO_x , CO and CO_2 emissions.

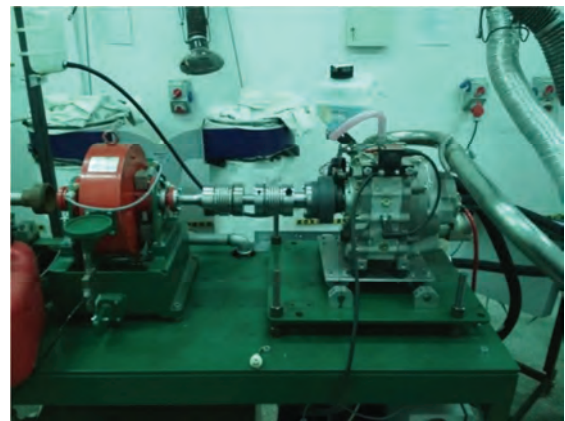


Fig. 5. The engine characteristics testing stand

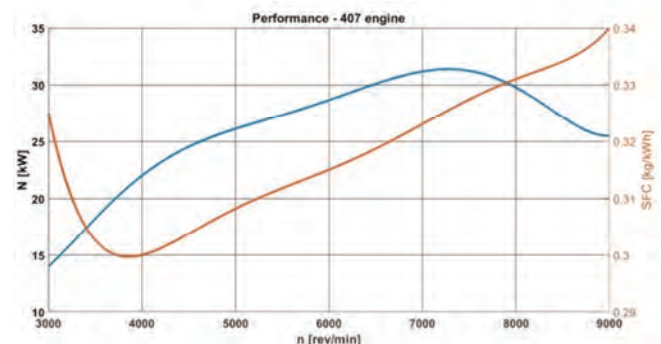


Fig. 6. Rotational characteristics of the 407 engine

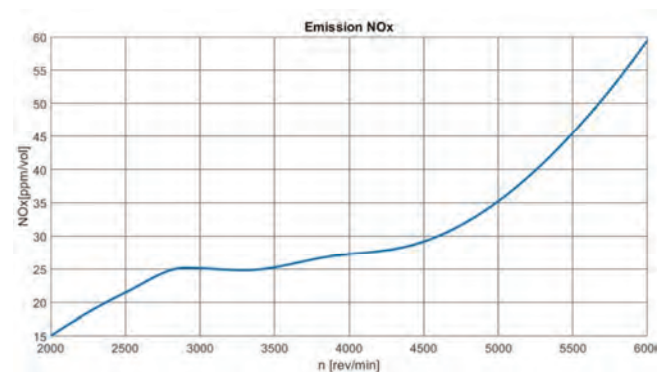


Fig. 7. The NO_x emission of the 407TGi engine

The analysis of the charts presented in the Figs 7–9, leads to the following conclusions:

1. The CO_2 and CO emissions were the lowest in the vicinity of the maximum torque obtained by the engines (about 4000 rpm), which at the same time was a point close to minimum of specific fuel consumption.

2. The NO_x emission grew continuously in the function of rotational speed, which was associated with the increase in the combustion temperature and the increase in fuel consumption by the engines [10, 13].

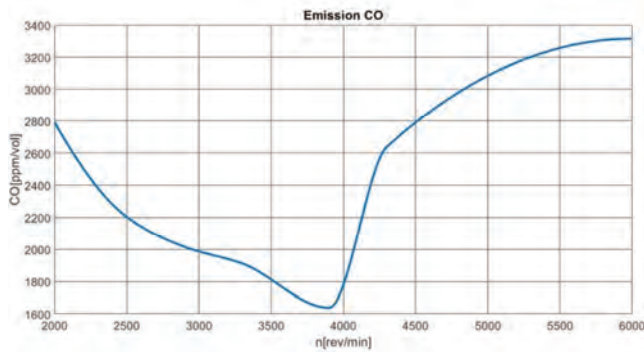


Fig. 8. The CO emission of the 407TGi engine

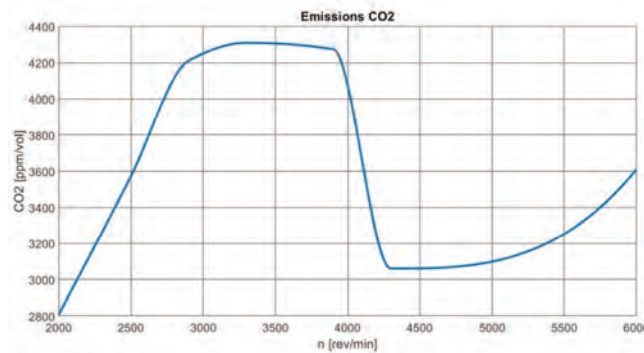


Fig. 9. The CO_2 emission of the 407TGi engine

The results obtained were used to conduct further research and analyses.

2. Selection of the distributed propulsion for the airframe

Taking into account the aerodynamic parameters of the airframe and the maximum power developed by the classic power unit, ten BLDC AXI 8120/10 engines were selected for the variant of distributed propulsion. The parameters of this engine are shown in Table 3.

Table 2. Technical data of AXI 8120/10[17]

Engine Type	BLDC	–
Maximum engine power	$N_{\max}$ [kW]	4.2
No load current	I_0 [A]	1.2
Maximum current	I_{IN} [A]	95
Maximum voltage	U_{IN} [V]	44.5
Engine pulse	K_v [RPM/V]	140
Voltage	R_M [Ω]	0.00466

Figure 10 presents a visualization of the distribution of engines installed on the motor glider.

Based on the technical data of the AXI 8120/10 [17] engine, its rotational characteristics were determined for the selection of propellers (Fig. 11). For this purpose, the formulas (1),(2),(3) were used to determine the input and output power (received on the shaft) of the engine. In this case the motor voltage control is assumed.



Fig. 10. Visualization of the propulsion concept

$$N_{IN} = I_{IN} \cdot U_{IN} \quad (1)$$

where N_{IN} – engine input power [W], I_{IN} – input current [A], U_{IN} – input voltage[V].

$$N_{OUT} = (I_{IN} - I_0) \cdot (U_{IN} - I_{IN} \cdot R_M) \quad (2)$$

where N_{OUT} – engine output power[W].

$$n = K_v \cdot (U_{IN} - I_{IN} \cdot R_M) \quad (3)$$

where n – engine revolution [rev/min].

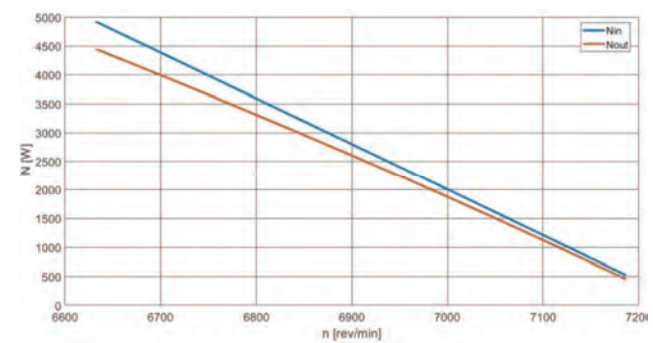


Fig. 11. The rotational characteristics of the AXI 8120/10 engine

Based on the calculated performance of the electric engine, using the available characteristics of the propeller [4, 8, 14] a two-blade propeller with a Clark Y2 profile and a diameter of 220 mm was selected. Such propellers allowed to determine the thrust of the power unit [4, 8] as the function of the flight speed of the motor glider. This is shown in Fig. 12.

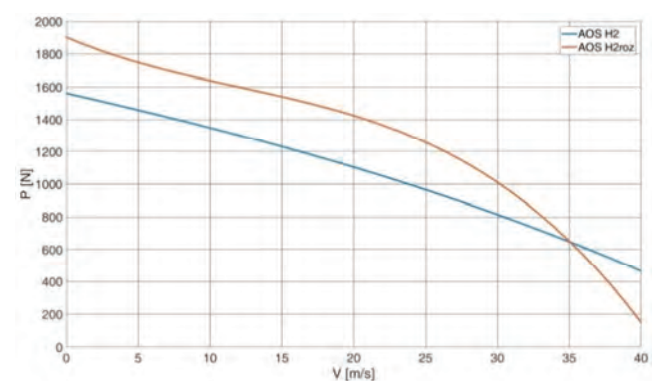


Fig. 12. Comparison of the thrust generated by distributed (red) and classical (blue) propulsions

This graph was used to determine the performance of the motor glider for both power units and for their comparison.

3. Comparison of the obtained power units performances and pollutants emission

For the assumed mission profile (i.e. start and flight, set at the altitude of 600 m, as illustrated in Fig. 13), the emission of individual compounds generated on a given route in the control volume was determined, assuming:

- The range until the energy source is depleted,
- The ambient conditions corresponding to the parameters according to the International Standard Atmosphere at a given altitude and the lack of wind.

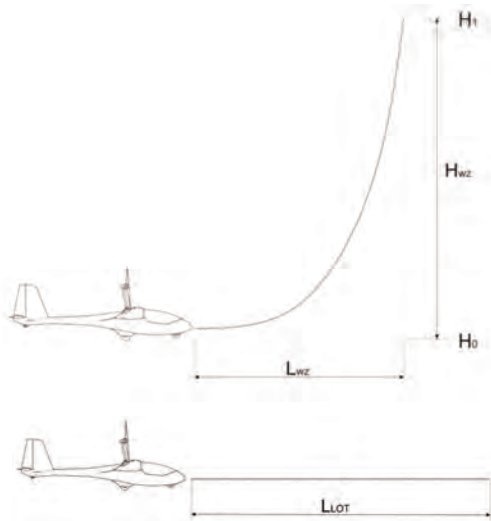


Fig. 13. Flight profile – climb (up) and horizontal flight (down)

On the basis of the chart presented in Fig. 12 and the aerodynamic data, it can be concluded that the classical propulsion, taking into account the efficiency of the propeller, consumes about 9 kW from the power source to generate thrust required for the flight. Under the same flight conditions, the distributed propulsion needs about 80% of this power to generate the same thrust.

Using the formula (4), the energy required for an hour of flight for both types of propulsions was determined:

$$E_h = x \cdot (N_n \cdot t_h) \div \eta_{sm} \quad (4)$$

where: t_h – flight time (1 h), N_n – power required for flight, x – number of engines (10 distributed, 1 classical).

Based on the formula (5), the energy stored on-board was determined:

$$E = I \cdot 3600[s] \cdot U_{bat} + \eta_{gen} \cdot (N_s \cdot t_s) \quad (5)$$

where: t_s – generator operation time [1]:

$$t_s = \frac{m_{fuel}}{SFC \cdot N_s} \quad (6)$$

N_s – power of rotary engine, SFC – specific fuel consumption 407 Tgi engine, m_{fuel} – mass of fuel stored on-board the aircraft (7 kg).

The total flight time was determined by:

$$t_L = E/E_h \quad (7)$$

Based on the calculations, for both drives there were determined the flight time and the corresponding range for a flight speed equal to 100 km/h. The results are shown in Figs 14 and 15.

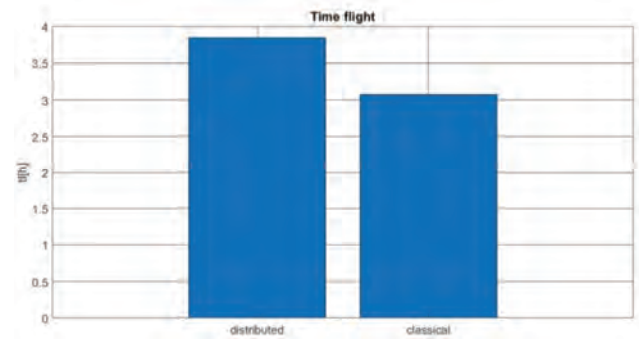


Fig. 14. The flight time

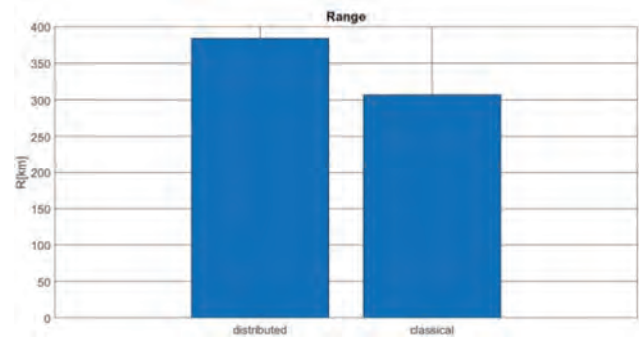


Fig. 15. The motor glider's range

The conducted research allowed to determine the emission for both propulsions. For comparative purposes, in Figs 16–18 the pollutants emissions during distance flight of 300 km are presented.

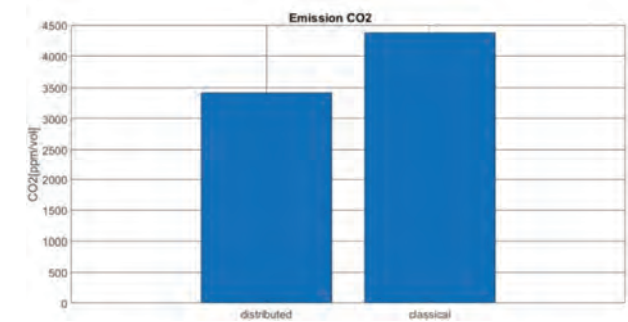


Fig. 16. The CO₂ emission

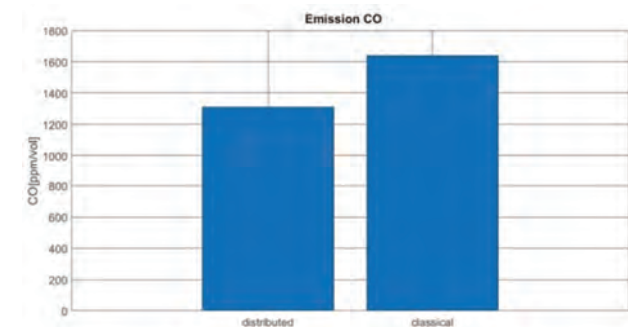
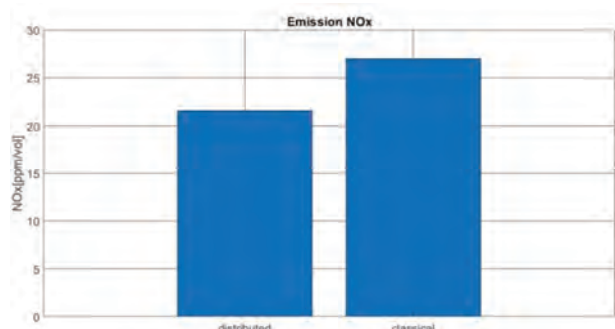


Fig. 17. The CO emission

Fig. 18. The NO_x emission

5. Conclusion

The analysis conducted leads to the following conclusions:

1. For the distributed propulsion, the range has increased by 76 kilometers and the flight duration by 46 minutes.

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Finite element analysis of a composite piston for a diesel aircraft engine

The article presents calculations of thermal and mechanical loads of the piston, consisting of two parts: steel and aluminum. The calculations were made using FEM in the Abaqus software. The piston is characterized by a split construction and was equipped with a cooling oil channel. The piston will be used in an aircraft diesel engine characterized by opposite piston movement. The presented geometry of the piston is the next of the ones being developed earlier and contains preliminary assumptions as to the size and main geometrical dimensions. The thermal boundary conditions of the simulation tests assumed defined areas of heat reception surface and heating of the piston by defining a temperature map on its crown. The results of these studies were presented in the form of temperature distribution and heat flux on the surface of the tested element. The strength boundary conditions assumed a mechanical load in the form of pressure resulting from the pressure in the combustion chamber applied to the piston crown surface and the opposite pressure defined on the support at the surface of contact between the piston and the piston pin. The results of these tests were presented in the form of stress distribution on the surface of the tested element. As a result of the analyses carried out, the results constituting the basis for further modernization of the piston geometry were obtained.

Key words: piston, combustion engine, FEM, thermal, mechanical loads

1. Introduction

The aviation industry has been recording steady and stable economic growth over the last eight years. Airlines achieve good financial results, generating over 30 billion dollars in profits per year. Operational efficiency has also improved. Flight load ratios in the last year increased to a record 81% [1]. The aircraft manufacturer, Boeing, assumes a 2.5-fold increase in the air traffic over the next 20 years. As a result, the industry will require more than 42,000 new aircraft up to 2037. It is anticipated that in 2019 commercial airlines will accept deliveries of over 1,780 new aircraft worth 80 billion USD. About half of the supplies will replace the existing fleet, which significantly contributes to the growth of fuel fleet efficiency [2].

The ultralight aircraft sector is also going in the same direction. Manufacturers of ultralight aircrafts and drones now expect new, light and economical engines with more power. However, the development of a new propulsion system requires numerous research and analyzes. It should be remembered that aircraft propulsion works in the atmosphere, and in very diverse conditions. Variable weather conditions (different pressure, temperature, oxygen level and moisture content) can change in a very short time and affect the engine's operation. The paper presents the problem of thermal load testing of the piston of the PZL-100 engine produced in cooperation with the Lublin University of Technology and WSK "PZL-Kalisz" S.A.

The PZL-100 is a new type of two-stroke internal combustion power unit using an opposing system of pistons placed in a single cylinder to create a combustion chamber. The charge is exchanged via the inlet and outlet ports of the cylinder alternately uncovering and closing the surface of the piston. The dosing of the fuel is via a dual common rail system powered by a central high pressure pump. In total, the engine has three cylinders and two crank and piston systems, independently for the inlet and outlet sides, synchronized with each other by means of a belt transmission. figure 1 shows the geometric model of the PZL-100 engine.

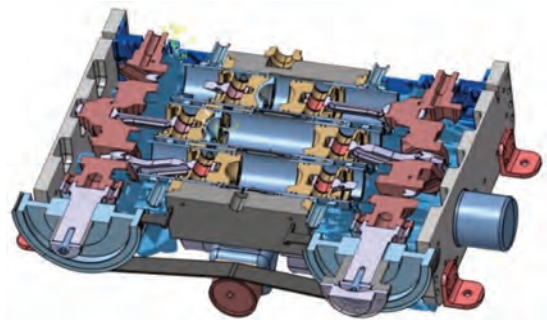


Fig. 1. Geometric model of the PZL-100 engine [3]

The key element of each engine is the piston. The thermal loading of the piston and the resulting temperature profile affect the operation of the drive unit. In diesel engines, the temperature profile in the combustion chamber is largely dependent on the number and orientation of injection holes, injection pressure, injection time, and the geometry of the piston crown [4, 5]. In the case of exceeded permissible of maximum temperature, loss of fatigue strength of the piston material and plastic changes may occur. The radial deformation of the piston causes changes in nominal dimensions and an increase in friction loss and looseness between the piston and other elements, which may cause plunger or contact with valves [6, 7].

Too high temperature in the area of the sealing ring may cause plastic deformation of the piston and increase wear of the sealer or its damage. In addition, overheating of the oil can lead to carbon deposits in the groove of the ring, which can act as a thermal insulator and cause sticking of the ring [8]. The newly developed engine design differs from the classic engine system with the valve system and the fuel injection along the cylinder axis. The combustion chamber in the PZL-100 engine consists of two piston heads. The fuel is injected perpendicularly to the cylinder axis from two injectors installed oppositely. All energy released during combustion process is transferred to the walls of the combustion

chamber. The numerical analysis of the thermal load on the piston crown and the temperature profile will make it possible to visualize the distribution of temperatures in key areas of the piston, such as the groove of the upper ring and the crown of the bottom. In the classic solution about 3–5% of fuel energy, in stationary combustion chambers, and 6–8%, in vortex chambers, is transferred to the piston. If the piston is not cooled, up to 60% of this heat can pass through the area of the piston ring to the cooling jacket [9]. Additional heat is passed through the bottom to the cooling jacket and from the bottom of the piston through the mist created as a result of the oil splash in the crankcase [10]. The intensity of walking the piston with oil can be increased by a spray jet. The interior of the crankcase is arranged with nozzles through which the oil is continuously sprayed onto the inside of the piston. This technology is the most widespread in gasoline engines and in diesel engines with lower power [11]. Pistons in the most modern racing engines and high-performance diesel engines currently use the design of internal cooling channels and chambers. In this case, the oil is injected through the injector into the central entry channel leading to the chamber located in the piston crown in close proximity to the piston crown. The coolant does not fully fill the chamber, but moves with an oscillating movement up and down with the force corresponding to the displacement of the piston during the reciprocating operation. This leads to a turbulent flow in the liquid cooling channel and intensifies the value of the heat transfer coefficient. The oil flows out of the chamber through channels arranged on the circumference of the piston directly to the crankcase. This type of solution ensures a more efficient flow of thermal energy from the area of exhaust gas interactions and provides a certain thermal insulation for the other elements of the piston. If the piston is cooled with oil, a significant part of this heat is discharged by the oil, reducing the relative amount that passes through the ring region [12, 13].

This paper presents the results of simulation tests of thermal and mechanical loads of the PZL-100 self-ignition two-stroke piston. The simulation tests carried out will be used to verify piston design and verify the new piston geometry developed specifically for this engine. The obtained temperature profile will expand the scientific base with new numerical models of heat loads of engine pistons with an opposing piston array with a combustion chamber between two piston heads and a double fuel injection perpendicular to the cylinder axis.

2. Geometrical model of the piston

The presented piston consisted of two parts: the upper made of steel 42CrMn04, the bottom of the aluminum alloy material M174 +. The Figure 2 shows the geometry of the piston.

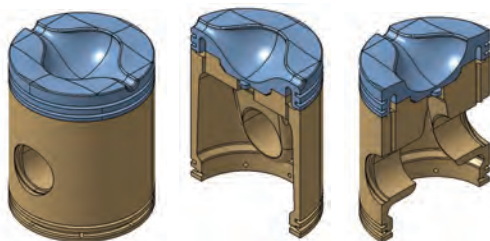


Fig. 2. Geometry of the piston

3. Boundary conditions

The research presented in the article was carried out in the Abaqus program using the Finite Element Method. Thermal and strength tests were performed separately, due to the simultaneous separate optimization work devoted to both issues. For both types of tests, the same computational grid was used. Only the properties of the grid were changed in terms of strength and thermal calculations. As demonstrated by preliminary simulation tests, the 1.1 mm element size is sufficient to obtain reliable calculation results. Increasing the density of the grid beyond this dimension does not significantly affect the calculation, and only causes an increase in their duration. The calculation grid is presented below. Such a geometry contained 394,102 computational elements in the lower part and 140,177 in the upper part. All elements were of the Tet type (Fig. 3).



Fig. 3. Calculation grid

The material properties of both parts are presented in the further part of the article. They were divided into strength and thermal properties.

3.1. Mechanical loads

Strength tests of the piston consisted in calculating the compressive strength of the piston, which was the result of the applied working pressure from the combustion chamber. The pressure value has been defined on the basis of the previously developed simulation models of engine operation [3].

Below, the boundary conditions that were given to the strength calculations of the piston are shown (Fig. 4). The piston model was constructed in such a way that the applied pressure on the piston's body balanced with the reaction pressure applied on the contact surface of the piston with the piston pin. In order to obtain a balance of forces, pressure was applied to the piston head, up to the contact surface in the first piston ring, the vertical component of the resultant force for this pressure was calculated and converted to the pressure applied to the contact surface of the piston with the piston pin. The pressures on these surfaces have been simulated in the form of a sinusoidal function, reflecting their actual distribution in the case of sliding contact. This mapping of piston working conditions allows a full analysis of the strength in the case of compression.

Defined loads on both sides of the piston allow its stabilization in three-dimensional space during simulation. However, because the calculated grid is not fully symmetrical, this introduces some imbalance of the element being tested. In order to remove unnecessary components of force-

es generated from pressures given, it was decided to confirm the piston in an additional place (on the opposite side of the bottom). The place where additional bracing was applied does not affect the results of the simulation.

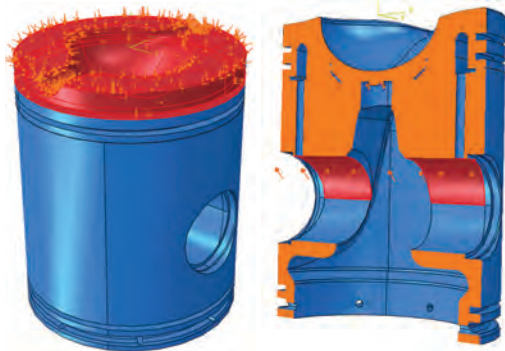


Fig. 4. Surfaces for which the following conditions were given: combustion pressure (left), reaction pressure from the piston pin (right)

The material properties adopted for calculations are presented in the Tables 1 and 2.

Table 1. Material properties accepted for calculation for material M174 +

Parameter	Unit	Value
Poisson's Ratio	–	0.33
Density	kg/m ³	2750
Young's Modulus	GPa	85

Table 2. Material properties accepted for calculation for material 42CrMn04+

Parameter	Unit	Value
Poisson's Ratio	–	0.3
Density	kg/m ³	7700
Young's Modulus	GPa	210

3.2. Thermal loads

Because the piston consists of two independent parts, it was necessary to give a coefficient of thermal conductivity occurring on the contact surface of two different materials (Fig. 5). The properties of this contact are shown in the Table 3, and the common (contact) surfaces have been connected via the Tie function [3].

Table 3. Heat flow conditions between surfaces [3]

Thermal conductivity coefficient λ [W/m K]	Value of the gap between surfaces [m]
700	0
0	0.001

Table 4 and 5 shows additional material data for thermal calculations for both materials used.

Table 4. Material properties of aluminium alloy called M174 + [1]

Temperature [K]	Heat transfer coefficient [W/m ² K]
293	130
623	140

Table 5. Material properties of steel called 42CrMn04 [1]

Temperature [K]	Heat transfer coefficient [W/m ² K]
293	33.5
523	34.0
773	34.2

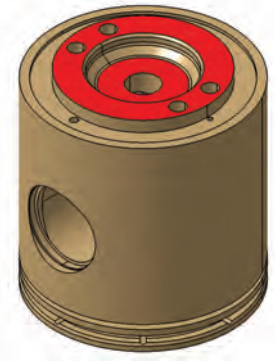


Fig. 5. Contact surface of two different materials

As boundary conditions, heat transfer coefficient map on the piston crown taken from CFD combustion model and constant piston crown temperature 850 K were assumed [3].

However, the following (Figs 6–9) shows the method of cooling the piston, which will probably be used in the actual version of the engine. Oil washing takes place via the cooling channel, but does not occur from the inside of the piston. Figures below shows the inner surface of the piston and the oil channel for which the boundary condition of the oil contact with the surfaces shown is given.



Fig. 6. The boundary condition of the piston's thermal load – side surfaces – temperature: 393 K, heat transfer coefficient: 500 W/(m²K)



Fig. 7. The boundary condition of the piston thermal load – bottom – temperature: 850 K, heat transfer coefficient: map

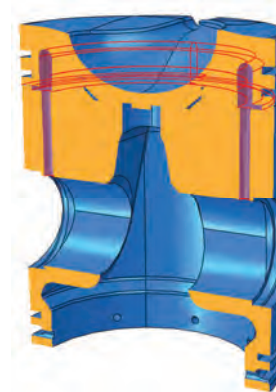


Fig. 8. Additional boundary condition of the piston's thermal load – oil channel – temperature: 373 K, heat transfer coefficient: 2500 W/(m²K)



Fig. 9. Additional boundary condition for the thermal load of the piston – internal surfaces – temperature: 393 K, heat transfer coefficient: 500 W/(m²K)

4. Results

4.1. Results of mechanical loads

Figure 10 presents the distribution of stresses on the surface and in the longitudinal sections of the piston. The distributions are marked with characteristic digits of the place. In order to improve the readability of the results, the scale was limited to 120 MPa. The maximum stress value is 250 MPa. The values in the range of 120–250 MPa are marked in gray. They are located inside the oil channel in the piston crown (1) and on the surface cooperating with the piston pin (2). The high value of stresses in the oil duct results from the fact that the outer part of the steel piston crown is not in contact with the aluminum bottom part of the piston. Between these parts there is a gap through which oil flows out. As a result, the outside of the bottom is bent. The high value of stresses in the second place results from

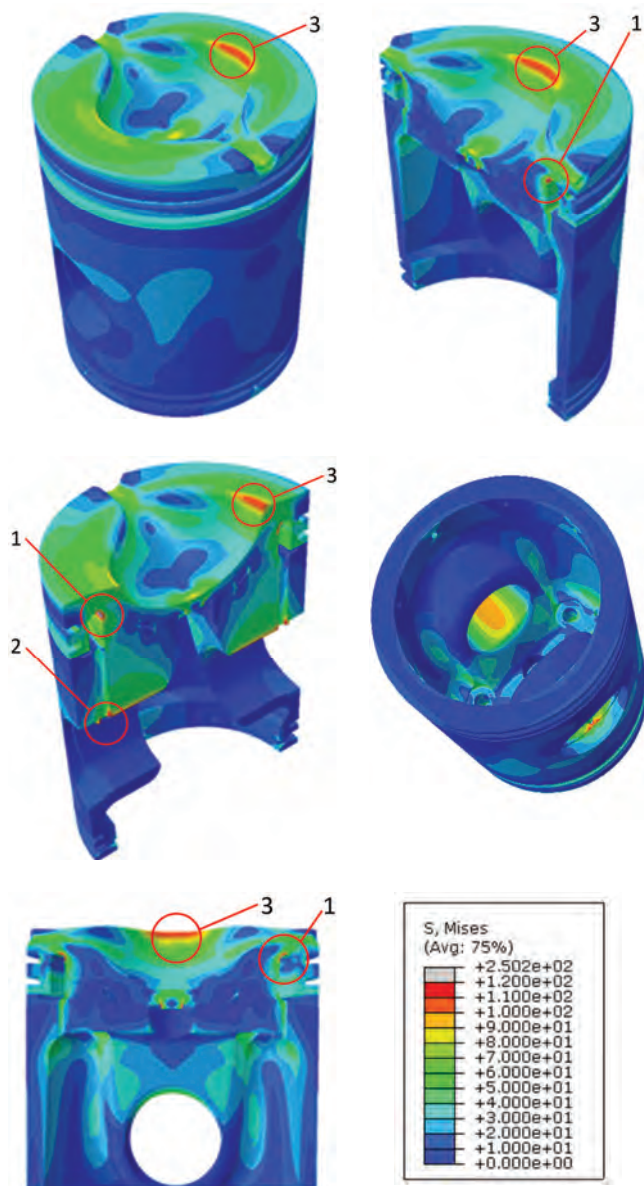


Fig. 10. Distribution of stresses on the surface of the piston

the pressure of the piston pin. Significant stress values were also observed at the edge of the combustion chamber (3). This is due to the bending of the piston crown, which is supported underneath by a piston pin.

4.2. Results of thermal loads

Figure 11 shows the temperature distribution on the surface and in the longitudinal sections of the piston. The maximum temperature was observed at the edge of the combustion chamber (T1) and it is 758 K. This location results from a small radius of edge rounding, from which the heat flow is hindered.

For the bottom part of the piston, made of aluminum alloy, the temperature reaches the value of 523 K (T2). The location of maximum temperature is located on the border of the piston division.

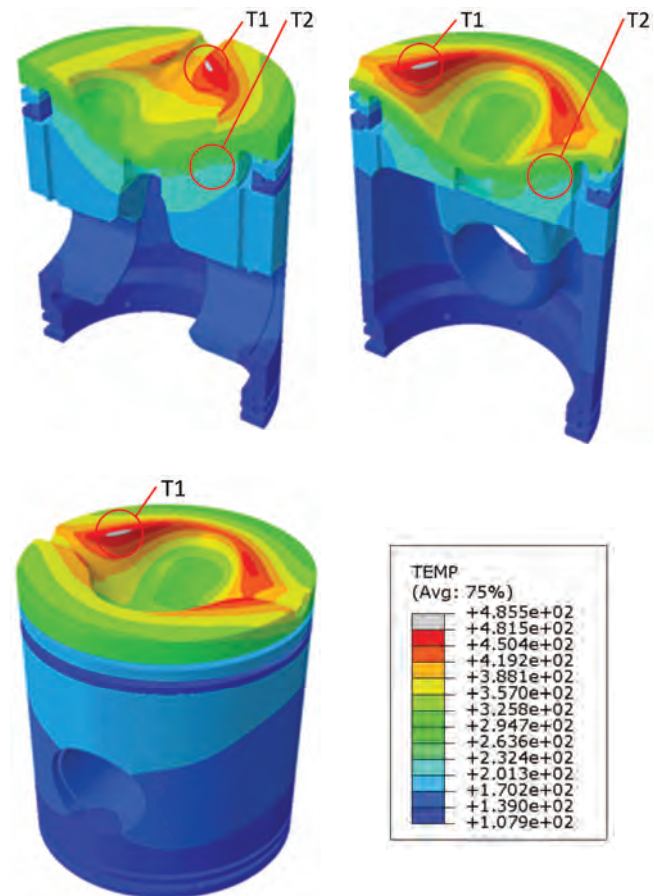


Fig. 11. Temperature distribution on the surface of the piston

Figure 12 shows the distribution of the density of the HFL heat flux flow on the surface and in the longitudinal sections of the piston. The maximum values were observed at the interface between the piston elements (H1). This is due to the accumulation of the heat flux that flows through the constrictions of the material with a small cross-section. Other characteristic places of occurrence of increased heat flow are recesses around the stream of injected fuel (H2) and inside the combustion chamber (H3). The reason for this location is the combination of a high temperature value and a small thickness of the piston crown.

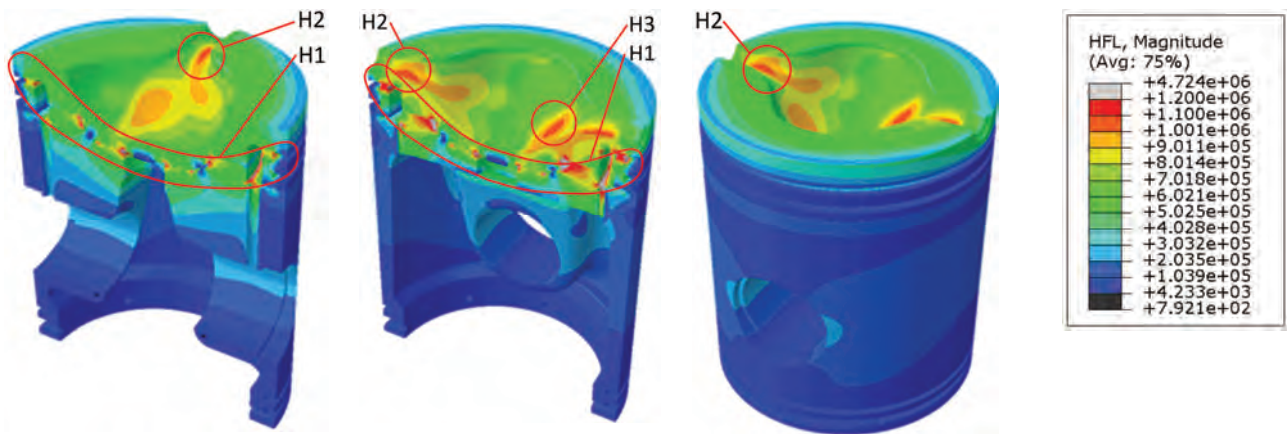


Fig. 12. Distribution of heat flow density on the surfaces of the piston

5. Summary

As a result of the analysis of the stress distribution, temperature and heat flux, it can be stated that the maximum values do not exceed the limit values for the materials used. The plunger in the pre-configured form will be made and used in the engine being constructed.

Acknowledgements

Acknowledgement: This work has been realized in the cooperation with The Construction Office of WSK "PZL-KALISZ" S.A." and is part of Grant Agreement No. POIR.01.02.00-00-0002/15 financed by the Polish National Centre for Research and Development.

Nomenclature

FEM finite element method
USD United States dollar

CFD computational fluid dynamics
Tet tetrahedronal

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Analysis of the impact of damage to the injection system on the emission of toxic substances in a spark-ignition internal combustion engine

The article presented analyses the impact of damage to the injection system on the emission of toxic substances in a spark-ignition internal combustion engine. The work focuses on the basic elements of the injection system, which include injectors, throttle position sensor, coolant temperature sensor and lambda sensor. In addition, a catalytic reactor has been included in the context of its direct cooperation with the injection system under set conditions. The toxicity of exhaust gases of different spark-ignition engines fueled with petrol or gas in determined operating conditions using the MAHA MGT5 exhaust gas analyser was tested. The content of toxic substances in the exhaust gases was recorded for the correct settings of the injection system and for the engine working with damage to this system. The tests carried out will allow to assess the impact of the damage of the injection system on the toxicity of exhaust gases.

Key words: spark-ignition internal combustion engine, exhaust gas toxicity, exhaust gas standards, injection system damage, exhaust gas analyzer

1. Introduction

An essential aspect of the emission standards is the proper condition of the internal combustion engine. Even the smallest damage to the fuel supply system can result in the fact that an engine approved for use and meeting the standards valid at the time it was produced will significantly exceed the emission thresholds. Consequently, damage to the injection system cannot be ignored by the car user. Any error message, or a visible symptom, i.e. power drop or uneven engine operation must be checked immediately for correctness and emissivity.

The content of toxic components in the exhaust depends not only on the engine's operating conditions, e.g. its thermal status, rotational speed, load. An important role is proper condition of the engine (e.g. the ignition advance angle, the correct operation of the exhaust gas recirculation system), and in particular the correctness of the mixture composition regulation. In MPI engines, dominating among spark-ignition engines operating on a mixture with a stoichiometric mixture, any change in the composition of the mixture results in a change in the amount of individual toxic components.

Another factor that affects the content of toxic components in exhaust gases is the technical condition of the exhaust gas treatment system – oxidation and reducing catalysts. Properly working exhaust after-treatment system – a three-way catalyst reduces the content of harmful HC, CO and NO_x combustion products in the exhaust. The composition of the exhaust depends not only on the quality of the combustion process taking place in the engine, but also on the technical condition of the components of the exhaust after-treatment system. For these reasons, the measurements of the toxicity composition of exhaust gases do not control whether the engine is ecological, but they are an important indicator of its technical condition [9, 10].

The measurements of the exhaust composition allow first of all for the general assessment of the correctness of

the combustion process in the engine, determination of the technical condition of the ignition and fuel system, the system of regulating the composition of the mixture and devices limiting the emission of harmful exhaust components. On the other hand, they do not allow to obtain a clear indication of the cause of the increase of toxic components emission. Therefore, diagnostic inference is based primarily on the analysis of the concentrations obtained for individual components and the knowledge and experience of the person performing the measurement.

If the combustion process in the engine is appropriate, the oxygen content in the exhaust gas should be within the range of 0.05–1%, the λ coefficient should be as close as possible to 1, the carbon dioxide content should be 14.5–16%, while the CO and HC (higher at idle speed than when the engine is running at high speed) should be smaller than the control values. For modern engines, the hydrocarbon content should not exceed 30 ppm. Measurement results differing from the above indicate a change in the ability of the engine.

For the tight exhaust system, the sum of the percentage of CO₂ and CO in the exhaust gas should be 15–16%. Significantly increased oxygen content in the exhaust gas, accompanied by a reduced concentration of carbon dioxide, indicates leakage of the exhaust system, and an increased content of hydrocarbons means a leak before the lambda sensor placed in front of the catalyst, which causes a malfunction of the lambda sensor and excessive enrichment of the mixture regulation [1].

The increased content of carbon monoxide, the reduced content of carbon dioxide and the high content of hydrocarbons indicate a poor thermal condition of the engine caused by, for example, damage to the coolant temperature sensor. This may indicate an incorrect combustion process. The reason may be improper operation of the mixture composition adjustment, damage to the lambda sensor or injectors, damage to the intake air temperature sensor, damage to the

air flowmeter or a permanently open valve in the circuit absorbing fuel vapors [4, 5].

A low CO_2 concentration and a low CO concentration below 0.1%, as well as a lambda value greater than 1.03, indicate a lean mixture. If there is a high content of hydrocarbons in the exhaust, this also indicates a leak between the piston and the sleeve. Defects in the ignition system (coils, spark plugs or wires) cause ignition failure – no fuel combustion.

A high hydrocarbon content of more than 30 ppm resulting in low carbon dioxide and high carbon monoxide is the result of catalyst malfunction.

The value of λ coefficient higher than 1.03, with a low content of CO and CO_2 as well as a high content of HC and O_2 , indicates that the mixture is too lean. The reason may be the lowering of the actual mass value of the sucked air indicated by the flowmeter due to the suction of additional air and leakages of the intake system behind the flow meter. The value of the coefficient λ lower than 0.97, with a high content of CO and HC and low CO_2 and O_2 , most often results from the excessive enrichment of the mixture caused by damage to the lambda sensor.

2. Analysis of the fuel system in spark-ignition engines

Fuel at a pressure of 1 bar in single-point SPI fuel injection is injected into the throttle area. The impact on the formation of a homogeneous fuel-air mixture has many factors. The most important is the high air velocity, thanks to which it is possible to mix the fuel with air. The conditioning factor is the air temperature above the throttle assembly. It can be heated not only by the heat emitted from the engine, but also by additional installation of engine coolant lines, or an electric heater.

The multi-point injection system is characterised by the fact that fuel is injected into the area of intake valves into a particular cylinder by means of an individual injector. This system of creating a mixture of fuel and air ensures identical supply of each cylinder. The working pressure on the injection bar is from 3 to 4 bar [6, 7].

In modern systems, the ECU (Electronic Control Unit) opens the injector synchronously when the valve is opened to a selected cylinder. The result of this is the creation of a homogeneous fuel-air mixture, which causes a significant reduction in the content of harmful substances in the exhaust.

3. Analysis of damage to injection system components affecting the toxicity of exhaust gases

The electric damages of the injector include: shorting the coil and interrupting the continuity of connection with the controller. Mechanical damages include: no fuel flow, no possibility of closing the needle valve due to contamination of the needle seat and lack of patency of the injector filter.

Possible faults associated with the coolant temperature sensor may be a shorting to the ground or open-circuit. The symptoms that accompany the damage may be: increased fuel consumption, increased idle speed or difficulty in starting the engine.

The lambda sensor acts as a feedback in the engine by measuring the amount of oxygen in the exhaust. The amount of oxygen measured is a reference to the air to fuel ratio in a mixture. If the composition of the mixture deviates from the value of $\lambda = 1$, the step change of the voltage signal sent by the sensor is immediately recognised and measured by the engine ECU. The signal sent by the lambda sensor varies from about 200 mV, meaning a lean mixture up to about 800 mV, meaning a rich mixture. The controller, based on the correction factor, changes the injection time, which regulates the proportions of the mixture. In this way, a cyclic change process is created that oscillates around the value of $\lambda = 1$ [2, 3].

The regulation of the composition of the mixture is carried out on an ongoing basis. This is a requirement for proper operation of the catalytic converter regarding removal of toxic substances from exhaust gases. The correct operation of the catalyst depends on the temperature above 350°C. Therefore, the composition of the mixture is regulated only after the exhaust system reaches a certain temperature. The correctness of the lambda sensor operation is monitored by the controller, and the faults are shown by MIL [8].

4. Experimental research

The aim of the analysis was to test the BMW M10B18 engine with spark ignition (Figure 1) to determine the composition of the emissions of individual exhaust components. The measurements were carried out for proper condition engine and simulated damage. Damage concerned the injection system and the sensors that affect the injection control parameters. Engine tests were performed three times in each damage. The emission of toxic substances was registered with the MAHA MGT5 analyser (Fig. 2).

Emissions tests carried out included:

- Determination of the carbon monoxide content in the exhaust gas of a BMW engine fueled with petrol and in the without ignition in individual cylinders (Fig. 3),
- Determination of the carbon monoxide content in the exhaust gas of a BMW engine fueled with petrol and for simulated damage (Fig. 4),
- Determination of the content of carbon monoxide in the exhaust gas of the BMW engine fueled with LPG and for simulated damage (Fig. 5),
- Determination of the carbon dioxide content in the exhaust gas of a BMW engine fueled with petrol and in the without ignition in individual cylinders (Fig. 6),
- Determination of the carbon dioxide content in the exhaust gas of a BMW engine fueled with petrol for simulated damage (Fig. 7),
- Determination of the content of carbon dioxide in the exhaust gas of a BMW engine fueled with LPG and for simulated damage (Fig. 8),
- Determination of hydrocarbon content in the exhaust gas of a BMW engine fueled with petrol and in the without ignition in individual cylinders (Fig. 9),
- Determination of hydrocarbon content in the exhaust gas of a BMW engine fueled with petrol and for simulated damage (Fig. 10),

- Determination of the hydrocarbon content in the exhaust gas of a BMW engine fueled with LPG and for simulated damage (Fig. 11),
- Determination of the content of nitrogen oxides in the exhaust gas of a BMW engine fueled with petrol in the without ignition in individual cylinders (Fig. 12),
- Determination of the content of nitrogen oxides in the exhaust gas of a BMW engine fueled with petrol and for simulated damage (Fig. 13),
- Determination of the content of nitrogen oxides in the exhaust gas of the BMW engine fueled with LPG and for simulated damage (Fig. 14),



Fig. 1. Test stand with BMW M10B18 engine



Fig. 2. MAHA MGT 5 exhaust gas analyzer

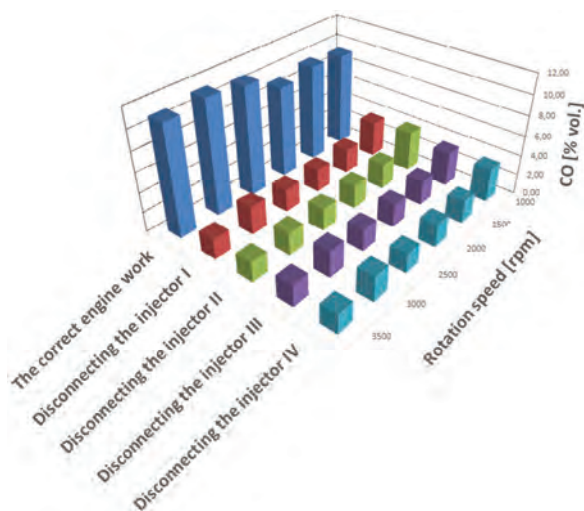


Fig. 3. The content of carbon monoxide in the exhaust gas of a BMW engine fueled with petrol and in the without ignition in individual cylinders

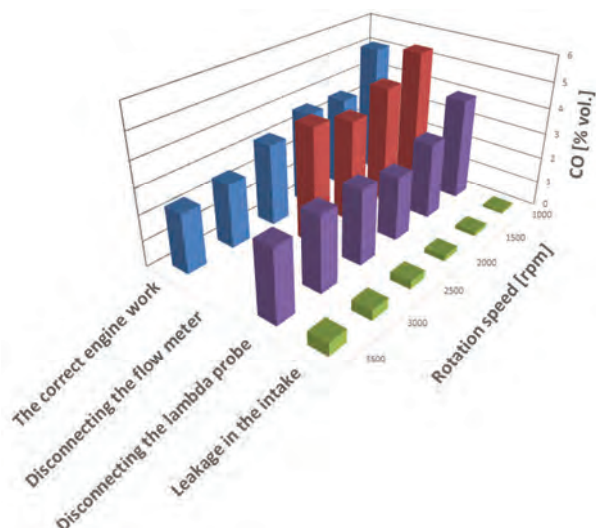


Fig. 4. The content of carbon monoxide in the exhaust gas of a BMW engine fueled with petrol working efficiently and for simulated damage

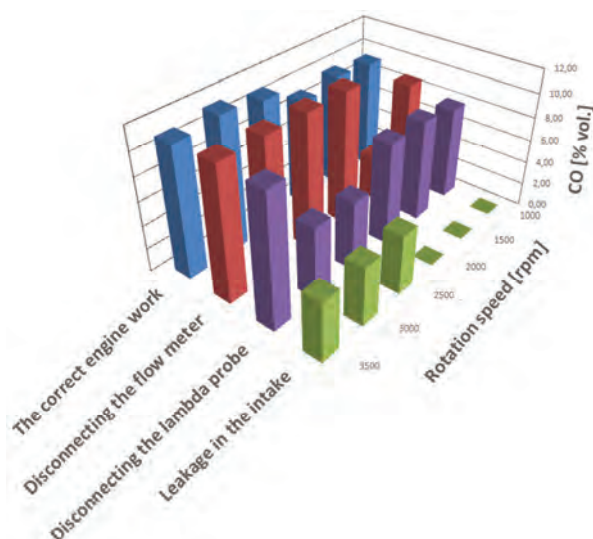


Fig. 5. The content of carbon monoxide in the exhaust gas of a BMW engine fueled with LPG working efficiently and for simulated damage

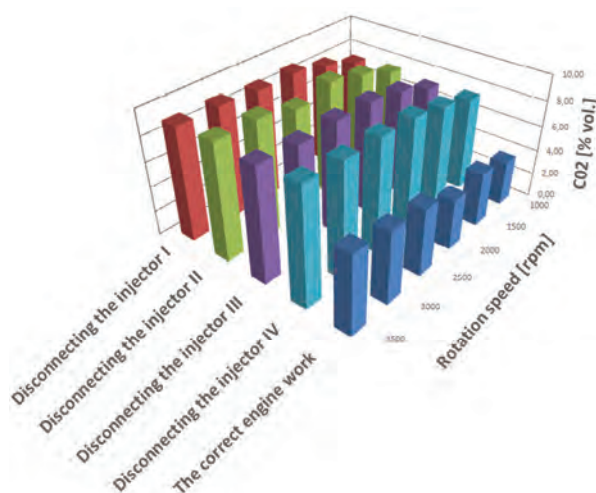


Fig. 6. Carbon dioxide content in the exhaust gas of a BMW engine fueled with petrol and in the without ignition in individual cylinders

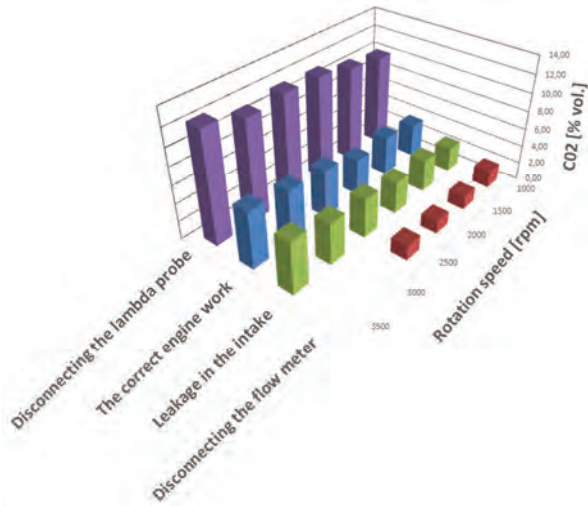


Fig. 7. The carbon dioxide content in the exhaust gas of a BMW engine fueled with petrol working efficiently and for simulated damage

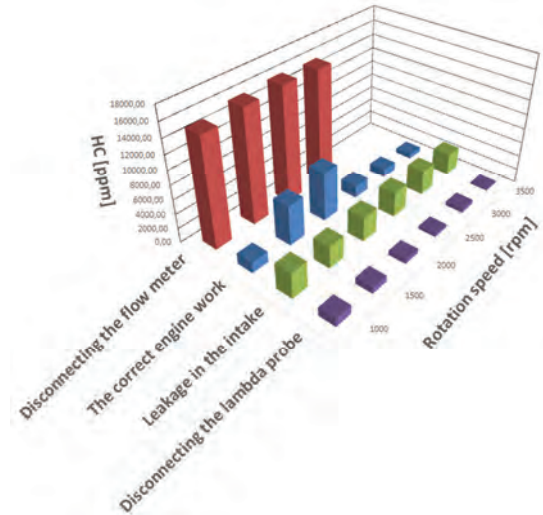


Fig. 10. The content of hydrocarbons in the exhaust gas of a BMW engine fueled with petrol working efficiently and for simulated damage

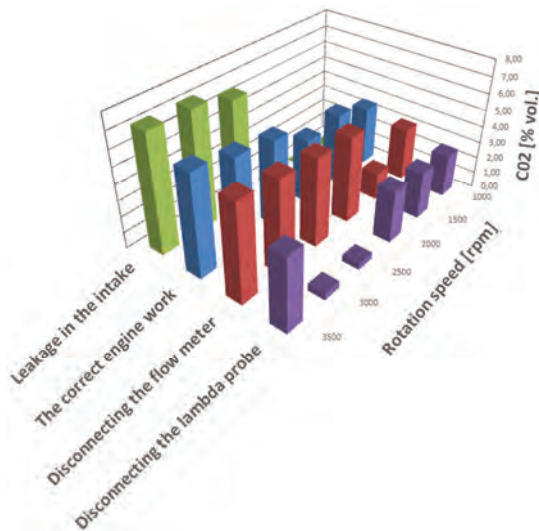


Fig. 8. The carbon dioxide content in the exhaust gas of a BMW engine fueled with LPG working efficiently and for simulated damage

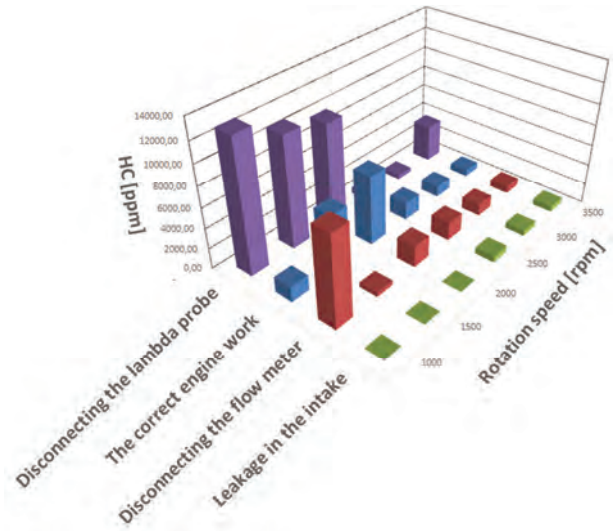


Fig. 11. The content of hydrocarbons in the exhaust gas of a BMW engine fueled with LPG working efficiently and for simulated damage

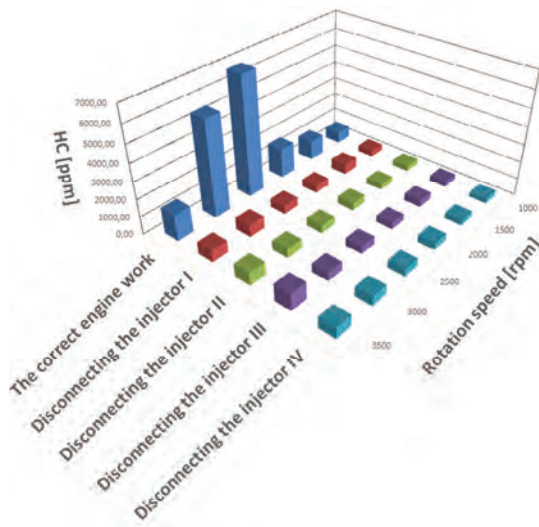


Fig. 9. The hydrocarbon content in the exhaust gas of a BMW engine fueled with petrol working efficiently and in the without ignition in individual cylinders

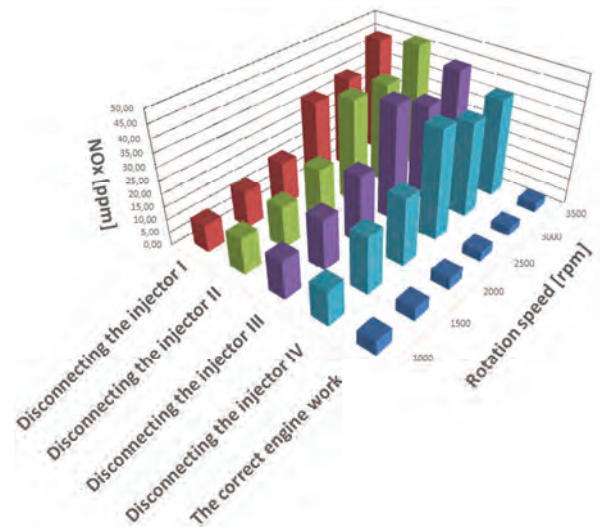


Fig. 12. The content of nitrogen oxides in the exhaust gas of a BMW engine fueled with petrol working efficiently and in the without ignition in individual cylinders

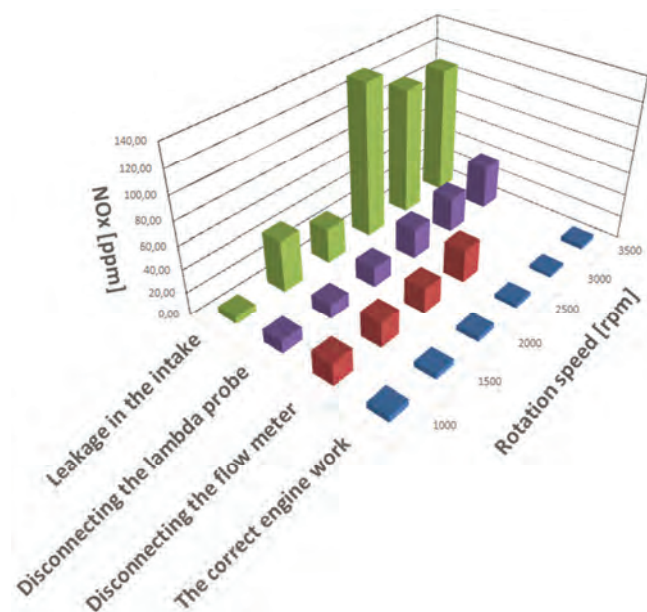


Fig. 13. The content of nitrogen oxides in the exhaust gas of a BMW engine fueled with petrol working efficiently and for simulated damage

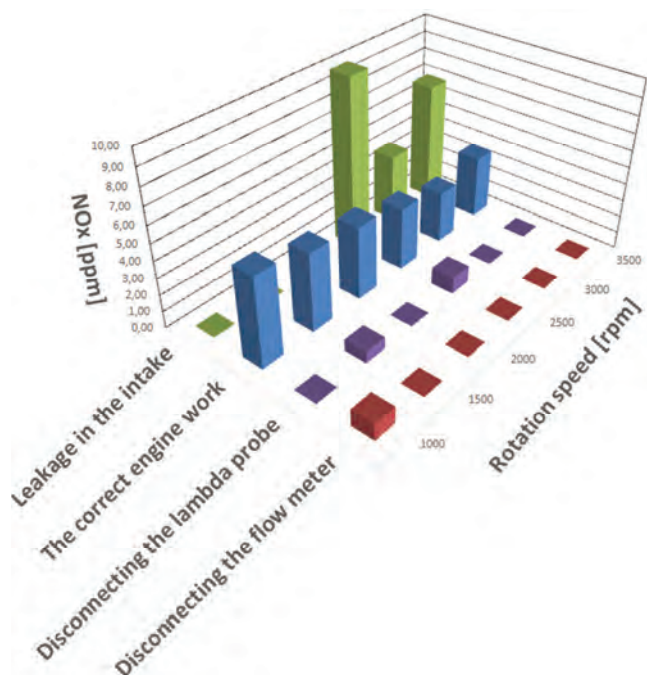


Fig. 14. The content of nitrogen oxides in the exhaust gas of a BMW engine fueled with LPG working efficiently and for simulated damage

Analysing the results of the injection system in the simulation of individual injector faults, the correspondence between the individual cylinders was found. In the case of the emission of nitrogen oxides, a slight increase is visible due to the impoverishment of the mixture to about 1.25-1.3. Concerning hydrocarbon emission, the decrease of their emission is visible at higher rotational speeds when an injector is damaged. An increase in emissions at lower engine speeds may be the result of engine damage. The decrease in the emission of carbon dioxide and carbon monoxide is the result of the without injection on single cylinder. Similar shapes of oxide emission runs and carbon

dioxide on individual cylinders show that each cylinder delivers a similar amount of combustion products.

The factors that increase exhaust emission have a damage of injection system. These factors include: incorrect signal from the air flow meter, disconnection of the lambda sensor or a leak in the intake system.

The emission of nitrogen oxides changes significantly in the case of leakage of the intake system. The increase of NO_x in relation to the correct operation of the engine is visible in the entire speed range.

The change in hydrocarbon emission is related to the leaking of the intake system and increases when the air flowmeter is disconnected. During the tests uneven operation of the engine occurred, causing the hydrocarbons to grow, which caused permanent damage to the measuring system.

The emission of carbon monoxide and carbon dioxide in the simulation of the air flow meter circuit caused the increase of carbon monoxide above the value at the correct engine operation. The carbon dioxide content in the exhaust gases assumes similar values with correct engine operation and disconnected sensor.

Increased concentration of nitrogen oxides in the case of fuelling the engine with LPG occurs when the intake system leaks at a speed above 2300 rpm (revolutions per minute). The reason for this behaviour may be a fuel and air mixture, which is leaked especially for higher rotational speeds. The decrease of nitrogen oxide emission in the case of a disconnection of the lambda sensor and the flowmeter results from the transition of the engine into an emergency mode of operation, which is connected with maintaining the mixture similar to stoichiometric.

Increased hydrocarbon emission values occur only in the speed range from 800 to 2300 rpm with a detachable lambda sensor.

The drop in the concentration of carbon dioxide in the event of the lambda sensor being disconnected and its increase in leakage of the intake system may be caused by a larger amount of sucked air.

Emission of carbon monoxide at rotational speeds from 800 to 2000 rpm is kept constant, after which it rapidly increases due to the impoverishment of the mixture and reaches its maximum value at 2,500 rpm. For higher speeds it maintains a constant value.

Analysing the surface plot of carbon dioxide emission for selected failures (Fig. 15), it can be concluded that the highest CO₂ values are obtained at the break in the lambda sensor circuit, and the smallest ones at the break in the flow meter circuit.

The hydrocarbon content (Fig. 16) is the highest at the flow meter circuit, while the smallest HC content occurs at the break in the oxygen sensor circuit.

The carbon dioxide content (Fig. 17) in the exhaust gases of a BMW engine fueled with LPG with an increase in speed for three failures increases as a function of the rotational speed. In the event of a break in the lambda sensor circuit for speeds between 2,500 and 3,500 rpm there is a sharp drop in the content of carbon dioxide.

Emission of hydrocarbons (Fig. 18) as a function of rotational speed reaches maximum values with a break in the

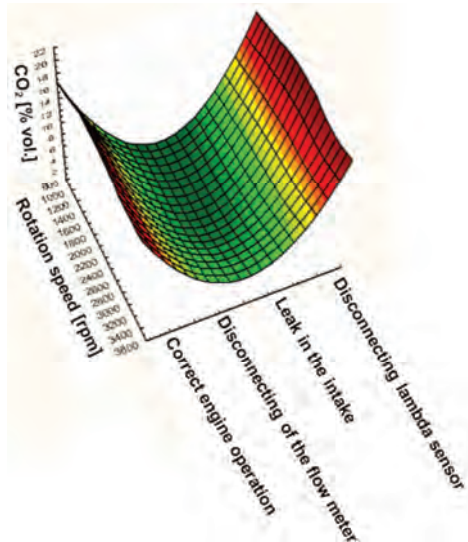


Fig. 15. The content of carbon dioxide as a function of rotational speed for an efficient engine and simulated damage – a petrol-fueled engine

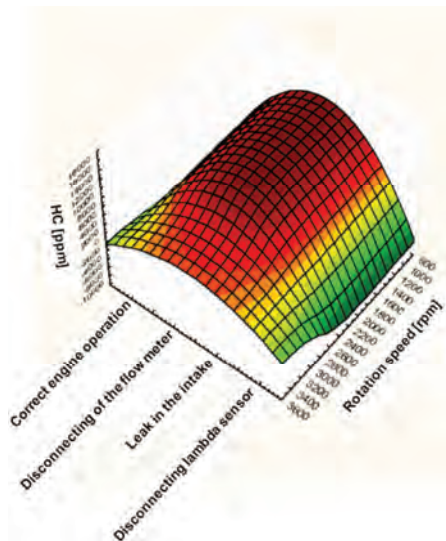


Fig. 16. The hydrocarbon content as a function of rotational speed for an efficient engine and simulated damage – a petrol-fueled engine

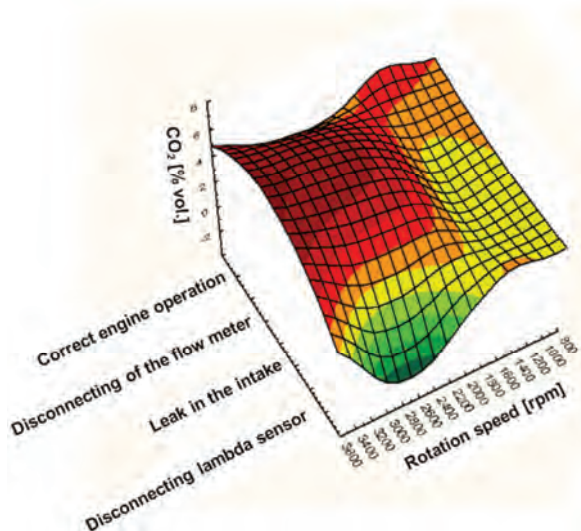


Fig. 17. Carbon dioxide content as a function of rotational speed for an efficient engine and simulated damage – an engine fueled with LPG

lambda sensor circuit at speeds from 800 to 1200 rpm. The smallest values are achieved when the flowmeter is disconnected and the intake leak is reached.

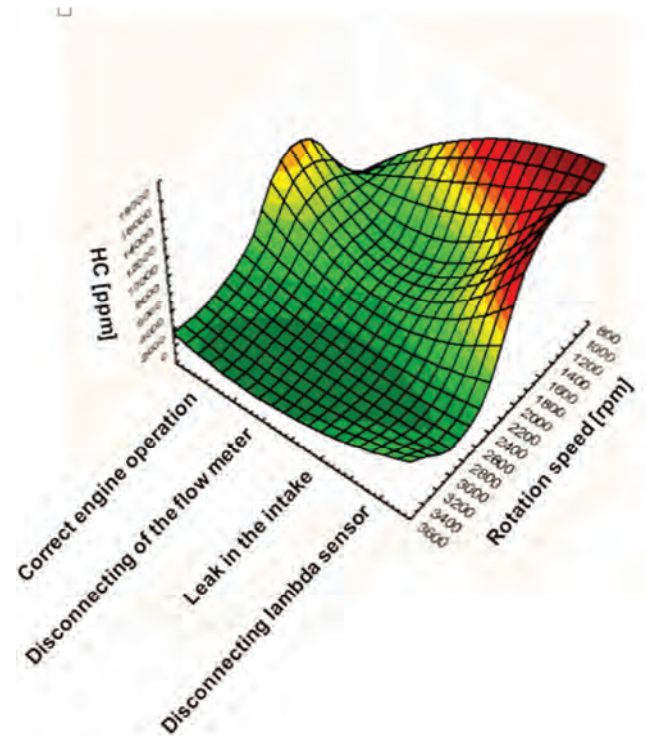


Fig. 18. The hydrocarbon content as a function of rotational speed for an efficient engine and simulated damage – an engine fueled with LPG

5. Summary

On the basis of the literature analysis and the tests conducted it can be stated that damage to selected elements of the injection system of a spark-ignition engine affects the combustion process.

Analysis of the measurements made for emission of individual components during fault simulation allows to determine the type of damage to the elements of the injection system. The tests carried out facilitate the diagnosis without interfering with the electrical installation. The use of the exhaust gas analyzer with the measuring apparatus allows to clearly determine the impact of the type of damage on the change of toxic exhaust components. The simulated damage to the flowmeter in the form of a break in the circuit, interruptions in the lambda sensor circuit and leakage in the air path caused the change of engine control, which changed the emission of toxic substances. Comparing the combustion process when switching off individual injectors it can be concluded that the combustion process on individual cylinders was correct taking into account the content of carbon monoxide, carbon dioxide, nitrogen oxides and hydrocarbons as a function of rotational speed, which were similar. The experimental studies carried out constitute input data for the development of algorithms for diagnostic patterns, based on the combination of error codes with the results of toxic components, which will allow to clearly determine the type of damage.

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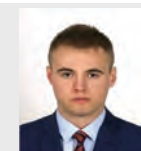
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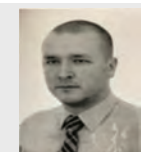
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Analysis of thermal parameters of heat storages for use in vehicles with combustion engines

The propulsion system of a vehicle using an internal combustion engine generates a significant amount of waste heat during operation, which is almost entirely discharged into the environment without any useful effect. One of the ways of using waste heat is storing it, and then using, for example, when starting the engine in winter conditions. The application of the indicated solution, in particular for the combat vehicle will allow to reduce the effects of cold start and will shorten the time of preparing such a vehicle for combat operations. The article presents: types of heat accumulators that could be used in a military vehicle, the results of preliminary tests carried out on the test stand and the impact of an additional heat source on the time of heating the internal combustion engine and on emission of exhaust gas components.

Key words: thermal energy storage, heat storage, waste heat energy, engine's heat balance, PCM

1. Introduction

The combat vehicle is designed to work in specific conditions of battlefield. An important feature that should be possessed is the ability to quickly and easily integrate into activities and regardless of the climatic conditions, especially low temperatures or high humidity [1]. An important feature of modern vehicles is the possibility of thermal camouflage, which would make it impossible or difficult to detect and identify by infrared devices of enemy [2]. The use of a heat storage in the military vehicles can significantly reduce the time of vehicle preparation for battlefield operations, as well serve as a protection system. For example, if the radiator was damaged, it could allow the drive unit to be protected from overheating [1] and reduce thermal image of vehicle in infrared by reduction of the exhaust gases' temperature [2]. In other words heat storage layout is an interesting solution to use in combat vehicles, which can improve the heat management of such vehicles [1, 2].

2. Heat storage systems

Literature studies confirm that the subject of heat storage is still the subject of interest among research centres. Often, these solutions are dedicated to specific purposes, therefore their construction must meet a number of specific requirements. We can distinguish three basic methods of thermal energy storage [7]:

1. Sensible heat storage
2. Latent heat storage
3. Chemical transformation heat storage

The first method is the simplest, it relies on heating the heat storage material using its specific heat capacity [4]. It can be, for example, a liquid or a solid. Water, oils and liquid waxes are often used from liquids [7]. In mobile applications, it is usually a coolant of the engine.

Table 1 contains some examples of material used in sensible heat storages.

Second – latent heat storage – use of phase change energy is based on the use of a material that in the temperature range prevailing within the heat exchange system (combustion engine together with auxiliary systems – environment) can occur in at least two physical states, in solid

and liquid. During the change phase (solid-liquid or liquid-solid), the heat associated with the phase is stored or released [3, 7].

Table 1. Sensible heat storage materials and their thermal properties in 20°C [3]

No.	Material	Thermal capacity [kJ/kg K]	Thermal conductivity [W/m K]	Density [kg/m ³]	Energy density [kJ/m ³]
1	Aluminium	0.945	238.4	2700	2551.50
2	Copper	0.419	372	8300	3477.70
3	Graphite	0.609	155	2200	13339.80
4	Water	4.183	0.609	998.3	4175.88
5	Sandstone	0.710	1.8	2200	1562

Table 2. Phase change materials and their thermal properties [3]

No.	Phase Change Material (PCM)	Type	Melting point [°C]	Heat of fusion [kJ/kg]	Thermal conductivity [W/m K]
1	Polyglycol E600	Organic	22	127.2	0.189
2	Butyl stearate	Organic	19	140	0.21
3	Palmitic acid	Organic	57.8	185.4	0.162
4	Capric acid	Organic	32	152.7	0.153
5	KNO ₃ /NaNO ₃	Inorganic	220	100.7	0.56
6	CaCl ₂ ·6H ₂ O	Inorganic hydrates	29	187	0.53
7	LiNO ₃ /KNO ₃ / NaNO ₃	Inorganic eutectic mixture	121	310	0.52
8	Capric acid (65.12%) and lauric acid (34.88%)	Organic eutectic mixture	19.67	126	0.21

The main advantage of phase change materials (PCM) is their high energy density, almost constant melting / freezing point and a significant reduction of storage dimensions compared to solutions without phase change [3, 7]. Table 2 contains some examples of PCM.

The third method – the use of heat of chemical transformations takes advantage of the fact that heat is stored through endothermic processes, while the release of stored energy takes place through an exothermic process [7]. At present, costs and complex processes caused that this method has not found a wider application [7]. When selecting

PCM for a particular application, one need to pay attention to a number of aspects as well as the parameters:

- Material

Depending on the chemical composition of the phase-changing agent, the following are different [4, 5]:

- phase change temperature level,
- value of specific heat for the solid phase and for the liquid phase,
- value of enthalpy of phase change,
- thermal conductivity,
- thermal and chemical stability,
- etc.

In order to better illustrate the diversity of PCM in Fig. 1, groups of materials are classified by temperature and enthalpy of the phase transition.

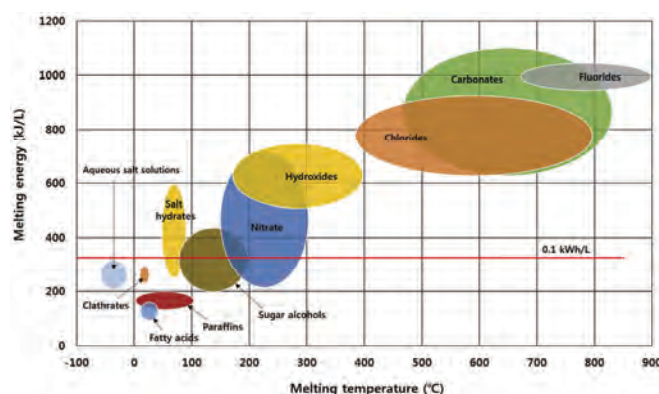


Fig. 1. Melting temp. and melting energy distribution of PCM [5]

- Working temperature

In the case of a vehicle with a combustion engine, the system with heat storage can be built in such a way that thermal energy is taken from the cooling system of the drive unit (temperature to approx. 90°C) or from the exhaust system (temperature up to 750°C), which differs depending on the size of the engine, its speed and load [6]. PCM should be chosen so that its melting point is in the temperature range of the system [5].

- Design of heat storage

The design of the heat accumulator, exchangers as well as the materials used for its construction are of great importance for the system performance. The thermal capacity of the storage, the heat transfer value with the environment and with the variable phase material, the PCM phase change time, the charging time, etc. will depend on this. [3]. In the literature, comparisons of materials used to build the storage can be found, from which it follows that apart from the conditions in which the heat storage system will work, it is also important to choose the materials of the storage elements for the PCM that will be in it [3]. Corrosion of materials is a particularly important aspect [3].

- Safety

The material should be non-flammable and non-toxic [4, 5].

- Availability

Ideally, the material should not expensive to buy and its availability on the market not limited in any way [4, 5].

- Destination

The purpose of the system and the way it is fitted are the issues for the designer who has to meet the project assumptions in an optimal way.

In mobile applications, up to now two methods have been used - the use of specific heat of the substance and / or the use of phase-change energy of the substance. In the first case, the storage is in the form of an insulated vessel with a channel system with a certain amount of coolant, which stores heat during operation [1], in the second case the storage has a special construction that allows intensification of heat exchange between the circulating liquid and the PCM [1]. The first designs of PCM storages sometimes resulted in leaks, which caused PCM to mix with the working medium which could result in environmental contamination [1]. PCM technology is under intensive development, constantly improved structures of storages are becoming more and more reliable. What's more, currently used substances are less inconvenient for the natural environment.

A comparison of the heat storage charging process based on glycol and PCM (SAT) was also made (Fig. 2).

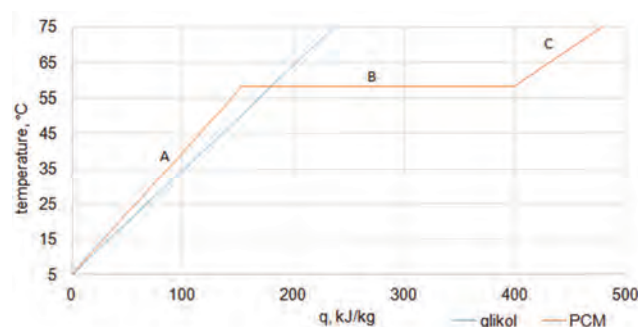


Fig. 2. Latent heat and sensible heat capacity (A – charging solid PCM; B – melting; C – charging liquid PCM)

From the Figure 2 it can be deduced that latent heat storage with SAT can store more energy in relation to sensible heat storage based on glycol (coolant).

3. Test of latent heat storage

3.1. Heat storage with sodium acetate trihydrate (SAT)

From a wide range of PCM materials, for the purpose of pre-tests was selected sodium acetate trihydrate ($\text{CH}_3\text{COONa} \cdot 3\text{H}_2\text{O}$) was selected. Properties of this substance allow it to be used in military vehicles. The most important parameters of SAT can be recalled from the safety data sheet and literature:

- not flammable [9]
- not explosive [9]
- not classified as hazardous or harmful to health and the environment [9]
- may irritate the eyes and upper respiratory tract (in case of high dust concentration in the air) [9]
- density (in liquid phase) 1280 kg/m³ [8]
- thermal conductivity (in liquid) 0.4 W/m K [8]
- heat of fusion 264 kJ/kg [8]
- melting point 58 °C [8].

On the test stand equipped with a heat storage containing 60 litres of SAT (Fig. 3), the temperature and flow of heat exchange factors were measured, in cycles - charging and discharging of the storage. The storage has two copper exchangers immersed in PCM, and from the outside it is insulated with PU foam and a PVC jacket. Charging of the storage was carried out with the help of an electric heater. A stand is equipped with a control panel and a set of sensors so it is possible to enter and monitor system operation parameters.

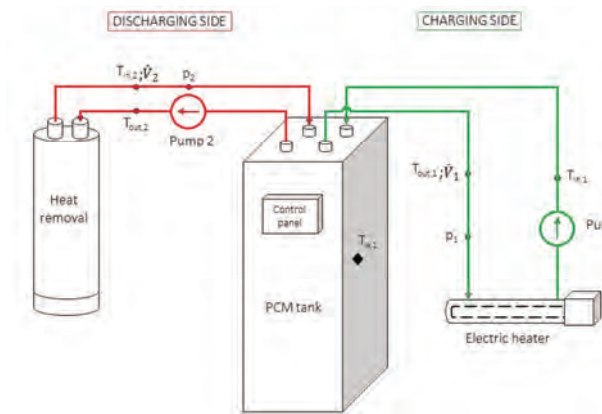


Fig. 3. Graphic scheme of the test stand at Institute of Thermal Technology, Silesian University of Technology, Gliwice

3.2. Exploration of test stand

Using an electric heater, the water was warmed up to 65°C and then a water stream 0.05 kg/s was charged in PCM in storage tank. Charging PCM took 3 hours. Next step was a discharge of the storage, which also lasted 3 hours. The water stream on the discharging side was also set to 0.05 kg/s. After the discharge, the tank was passively cooling itself during 5 days.

Based on the obtained test results, the characteristics of the loading process and discharging were made (Fig. 4). The temperature fluctuation which can be observed in the tank wall temperature diagram is caused by the fact that in one room there were also tests carried out at another stand. When working with the storage, the wall temperature sensor may have been exposed to the ambient temperature.

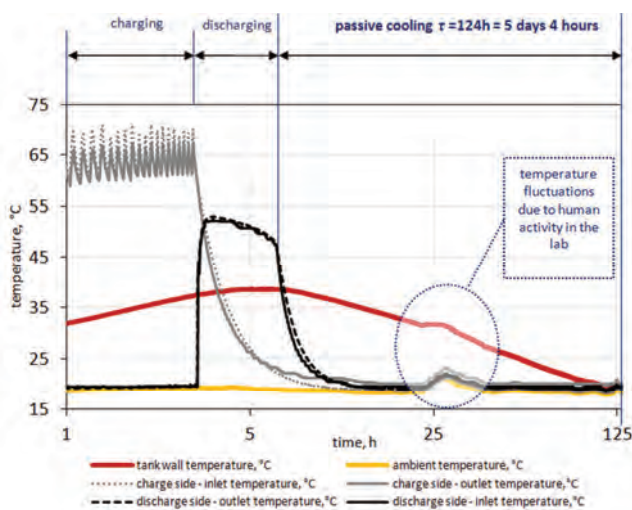


Fig. 4. Results of PCM tests

4. Engine's warm up process

4.1. Description of process

Analytical calculations of the process of self-heating of the CI engine under idle condition were carried out. The calculations were made for two variants with and without the use of physical enthalpy of the exhaust gases. In the first variant, a heat exchanger was used to partially recover the enthalpy of the exhaust gases, while in the second variant the exhaust gas was discharged directly to the surroundings. The exhaust gas warmed up the engine coolant, causing the internal combustion engine warm up faster. In order to determine the time needed to warm up the engine at ambient temperature, were used the results of experimental tests obtained earlier on the test stand. Scheme of test stand is presented in Fig. 5.

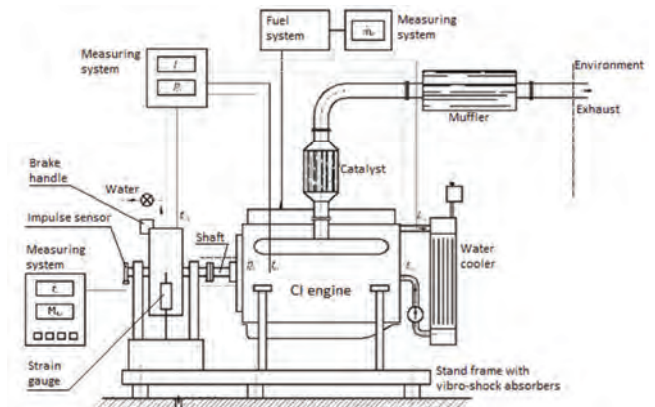


Fig. 5. Engine's test stand scheme [11]

Test stand is equipped with a 6-cylinder CI engine with an oxidizing catalyst. Engines parameters:

- volume: 6540 cm³,
- nominal power: 101.5 kW (2600 rpm),
- maximum torque: 423.65 Nm (1600 rpm).

4.2. Heat balance of the engine

The analysis of the heat up process was made using the first law of thermodynamics. The balance of energy includes the chemical and physical enthalpy of substrates and products of the combustion process. Gases were treated as semi-perfect taking into account the influence of temperature on the change in the specific heat of individual components. The heat flow delivered to the environment was calculated based on the dependencies resulting from the equations describing the heat flow. The average value of the heat transfer coefficient (as a total, i.e. convective and radiation) from the external elements of the engine, to the environment was determined based on criterion equations for free convection. Figures 6 and 7 show schemes of the energy balance of an internal combustion engine, during the heating of the engine with (Fig. 7) and without (Fig. 6) heat coming from the enthalpy of exhaust gas.

Engine's heat balance equation:

$$(\dot{I}_{chp} + \dot{I}_{fp} + \dot{I}_{fa})\Delta\tau = \Delta U_e + (\dot{I}_{che} + \dot{I}_{fe} + \dot{Q}_{ot})\Delta\tau \quad (1)$$

Internal energy increase (engine & fluids):

$$\Delta U_e = \Delta U_g + \Delta U_o + \Delta U_s \quad (2)$$

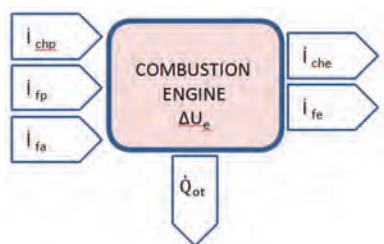


Fig. 6. Scheme of heat balance of the engine

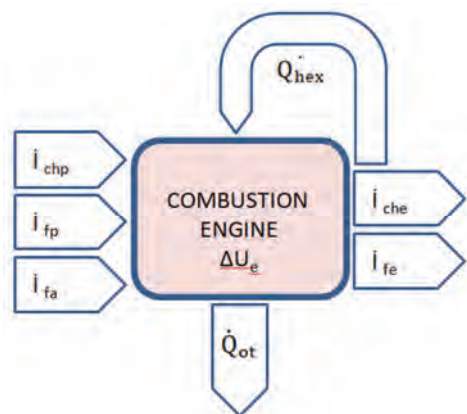


Fig. 7. Scheme of heat balance of the engine including heat coming from exhaust gas

Engine's heat balance equation:

$$(\dot{I}_{chp} + \dot{I}_{fp} + \dot{I}_{fa} + \dot{Q}_{hex})\Delta\tau = \Delta U_e + (\dot{I}_{che} + \dot{I}_{fe} + \dot{Q}_{ot})\Delta\tau \quad (3)$$

where: $\dot{I}_{chp}$ – chemical enthalpy of fuel [W], $\dot{I}_{fp}$ – physical enthalpy of fuel [W], $\dot{I}_{fa}$ – physical enthalpy of air [W], ΔU_e – internal energy increase [J], $\dot{I}_{che}$ – chemical enthalpy of exhaust [W], $\dot{I}_{fe}$ – physical enthalpy of exhaust [W], $\dot{Q}_{ot}$ – flux of heat loss [W], ΔU_g – energy of coolant [J], ΔU_o – energy of oil [J], ΔU_s – energy of steel [J], $\dot{Q}_{hex}$ – heat flux from heat exchanger [W], τ – time [s].

4.3. Assumptions and test data

As mentioned earlier, the analysis of the engine's heating process was carried out using the previously obtained experimental data. The data was obtained during tests of the engine operating in a steady state (quasi-stationary). Additionally, for the purposes of balance calculations:

- ambient temperature -30°C ,
- end temperature I $+30^\circ\text{C}$ – it was assumed that this is the temperature at which the vehicle can slowly move from the lowest gear positions,
- end temperature II $+65^\circ\text{C}$ – it was assumed that this is the temperature at which the vehicle can put maximum load on the drive system,
- variable engine speed: 600 rpm and 1200 rpm,
- variable engine load: idle, 90 Nm and 180 Nm.

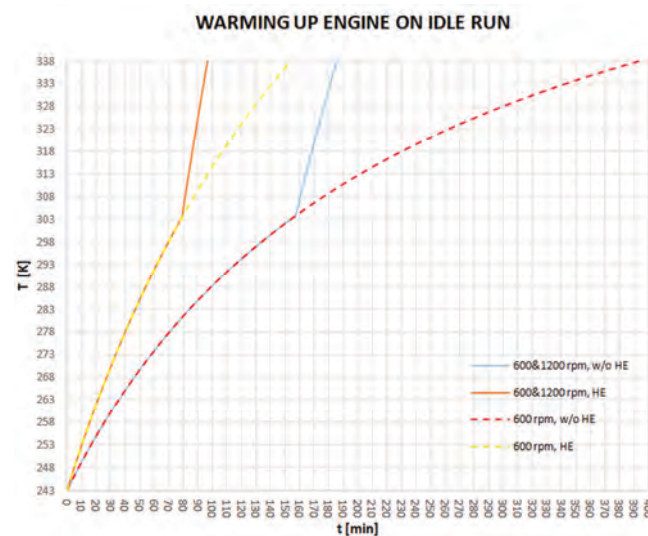
The stream of consumed fuel was measured using a mass flow meter.

During the test, the composition of dry exhaust gases in the form of volume shares of the following substances was measured: NO_x , CO, CO_2 , HC* (*equivalent to C_6H_{14} [10]).

In theoretical-experimental analysis, it was assumed that the average composition of the exhaust gases during the heating of the internal combustion engine will be similar to the data obtained in the stationary state during experimental investigations. In the future, it is planned to perform tests in a transient state taking into account the heating characteristics and the emission of harmful exhaust components.

4.4. Results

The measurements were made at engine speeds of 600 rpm and 1200 rpm without engine load (idle run). Figure 8 presents the characteristics based on measurements and calculations of thermal balances.


 Fig. 8. Warming up diesel engine diagrams from -30°C [243.15 K] (HE – with heat exchanger; w/o HE – without heat exchanger)

The engine warm-up from -30°C at 600 rpm under idle condition is shown using dashed lines. Red colour without additional heat source, and yellow with heat exchanger. In addition, after the engine reached $+30^\circ\text{C}$, the effect of increasing the speed of rotation to 1200 rpm on the heating time was checked (continuous lines of red and blue).

Figure 8 confirms that the effect of the additional heat source during engine start-up is significant. The engine warm up time, to $+30^\circ\text{C}$ at 600 rpm under idle condition has been reduced by 70 minutes and by 240 min in the case of heating up to $+65^\circ\text{C}$. Increasing the engine speed to 1200 rpm after reaching the temperature of $+30^\circ\text{C}$ caused that the heating times were further shortened. The results of theoretical calculations, showing the reduction of heating time (Fig. 8) may contain error due to the adopted, simplified heat exchange model.

4.5. Comparison with research in a climatic chamber

In order to verify the theoretical analysis presented above, the results from the test in a climatic chamber, a similar six-cylinder CI engine are shown. In Figure 9 the results of the temperature increase of engine parts and fluids measured during warming up process in climatic chamber are presented. At the beginning of the experiment the engine fluids has been preheated using combustion heater switched on for about 58 minutes. In the first part of the experiment the engine was operated under idle condition.

After 40 minutes of engine work the functional test of vehicle started which lasted two hours [12].

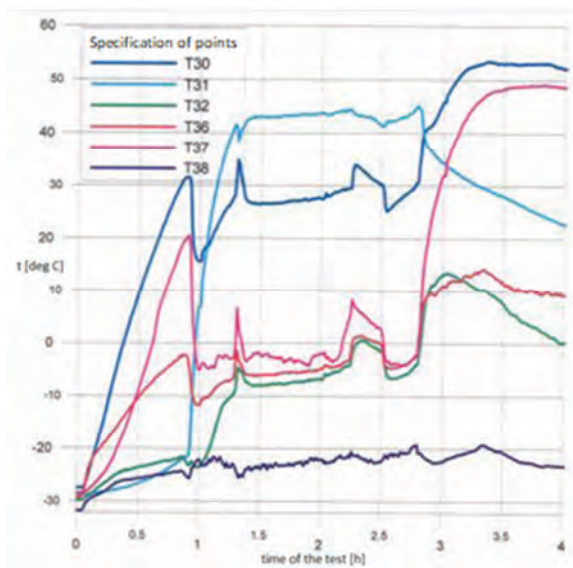


Fig. 9. The course of temperature change at selected points of the CI engine [12]

Description to Figure 9 [12]: T30 – temperature on the surface of the unit with the sensor to measure the temperature of the coolant, T31 – temperature of the engine oil in the oil sump, T36 – temperature on the surface of the engine block (sensor in the mounting – technological hole).

From the Figure 8 and 9 it can be seen that the difference in the results of the theoretical calculations and experimental test of similar engines is not significant. The reason of shorter time observed for experimental test is the use of combustion heater at the beginning of experiment.

4.6. Analysis of exhaust components

In addition to the analysis of the engine's heating time, the emission of harmful exhaust components was also analyzed (Fig. 10, 11, 12 and 13) based on the times shown in Fig. 8. It has been confirmed that the commissioning of a preheated engine significantly reduces the emission of harmful exhaust gas components.

Emission of harmful exhaust gas components was also carried out for the engine working under idle condition at 600 rpm and 90 Nm and 180 Nm load at 1200 rpm. Concentration of exhaust components are shown in Fig. 14 and Fig. 15.

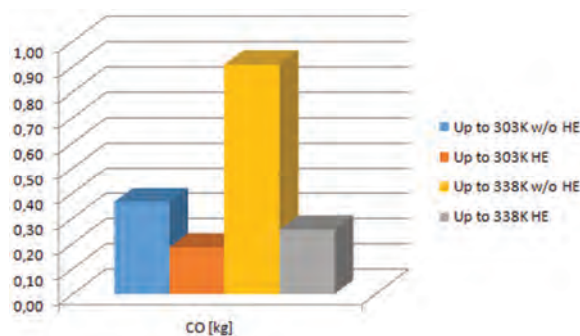


Fig. 10. Amount of CO after warming up (HE – with heat exchanger; w/o HE – without heat exchanger)

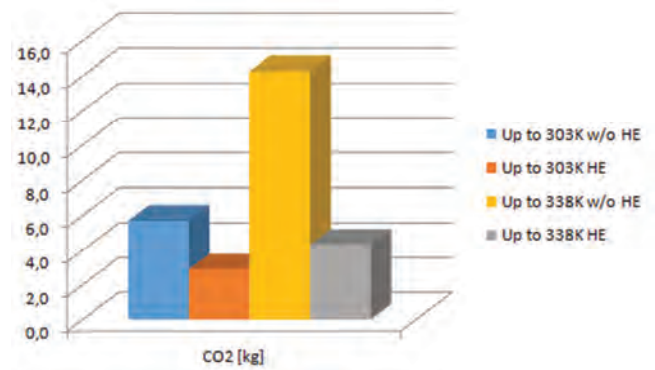


Fig. 11. Amount of CO₂ after warming up (HE – with heat exchanger; w/o HE – without heat exchanger)

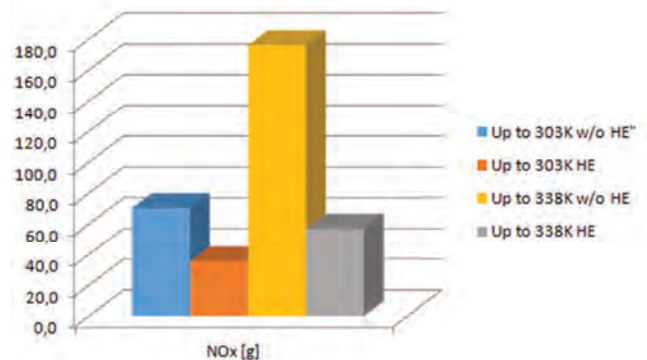


Fig. 12. Amount of NO_x after warming up (HE – with heat exchanger; w/o HE – without heat exchanger)

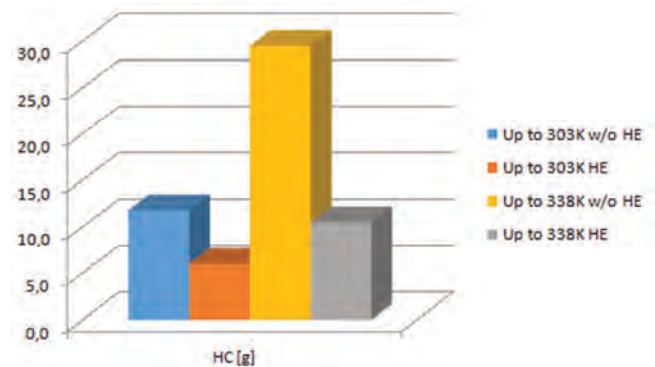


Fig. 13. Amount of HC after warming up (HE – with heat exchanger; w/o HE – without heat exchanger)

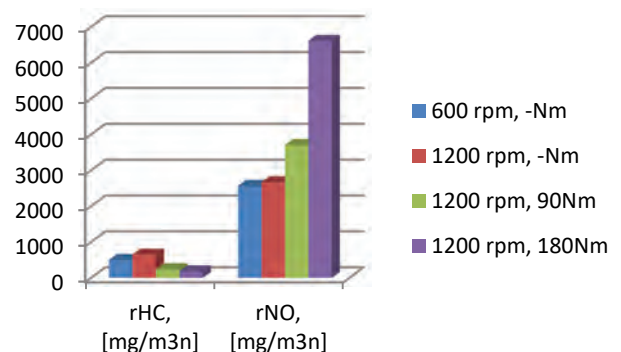


Fig. 14. Amount of NO_x and HC concentration in exhaust gases

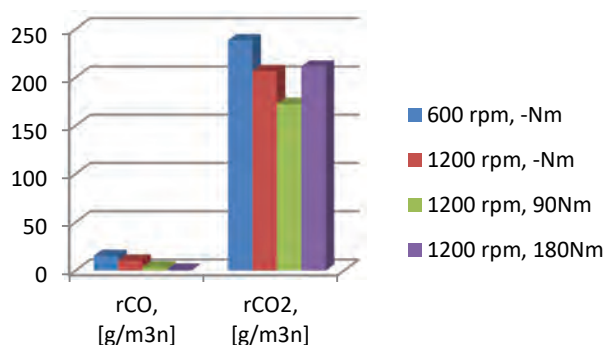


Fig. 15. Amount of CO and CO₂ concentration in exhaust gases

The high NO_x value in Fig. 14. is caused by the fact that the engine used only an oxidation catalyst.

4.7. Theoretical analysis of the engine temperature depending on the amount of SAT in the heat storage

Based on the previously held assumptions and using the energy balance equation recorded for the system: combustion engine – PCM (SAT) storage, has been prepared characteristics (Fig. 16) which presents temperatures to which

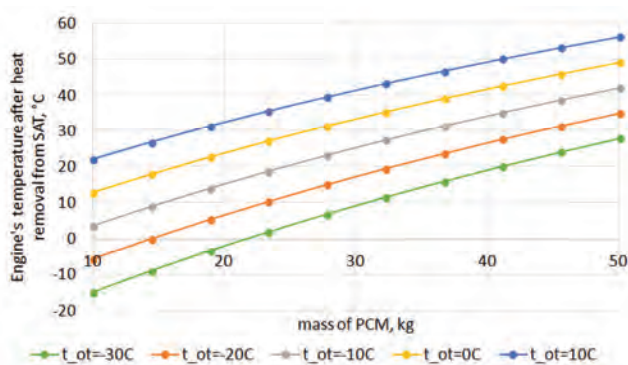


Fig. 16. Engine's temperature in relation to heat storage mass

the given mass of PCM material (assumed 10, 20, 30, 40 and 50 kg) can heat the motor body depending on the ambient temperature (t_{ot}). The range of adopted ambient temperature from -30°C to $+10^{\circ}\text{C}$.

5. Conclusions

The comparison of conventional heat storage system and the system with phase change material shows that the PCM system can achieve significantly less mass for the same heat capacity. The PCM accumulator, based on sodium acetate trihydrate has more than two times less mass comparing to the conventional accumulator based on glycol.

PCM (SAT) can be used in combat vehicles but due to the low melting point of SAT, it is not suitable for storing heat from the exhaust system. Therefore, reducing the thermal image of a vehicle in thermal imaging systems will not be effective.

The tests and analyses, carried out on the combustion engine and the heat accumulator have confirmed the validity of using heat storage systems in combat vehicles, which often have to be activated at low ambient temperatures in order to quickly and efficiently join to battlefield activities.

Analytical calculations of the process of self-heating of the CI engine confirms, that the additional heat source during engine start-up, significant reduce engine's heating time and emission of toxic exhaust components to the environment. To confirm the obtained results, it is necessary to perform experimental tests on the vehicle during natural conditions.

The design of the heat accumulator should be adapted to the vehicle. A well-chosen heat storage's capacity will allow to warm up the engine parts and fluids to the desired temperature within a certain time.

Selection of PCM and the design of the heat storage system for the combat vehicle, requires further R&D works.

Nomenclature

PCM phase change material
SAT sodium acetate trihydrate

CI compression ignition

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Pyrolysis oil combustion in the CI engine

Pyrolysis oil obtained from thermal biomass processing (torrefaction and pyrolysis) was used as an additional fuel for the compression-ignition engine equipped with a classic (non-common rail) injection system. The basic fuel used to the engine was regular diesel fuel. The tests were carried out with two content of pyrolysis oil in diesel fuel as follows: 10 and 20% by volume. In addition, the combustion process was investigated in the engine operating only on pyrolysis oil. The test results were based on a comparative analysis, where the diesel fuel was used as the reference fuel. The obtained results indicate that is a real possibility of co-combustion of pyrolysis oil with diesel fuel in the CI engine. On the other hand, a decrease in engine power resulting from the lower calorific value of pyrolysis oil and a greater unrepeatability of engine consecutive work cycles were observed.

Key words: bio-oil, pyrolysis, CI engine

1. Introduction

Nowadays, bio-oil is known as a black/grey liquid obtained from used tires or biomass from their thermal processing called pyrolysis/gasification. As reported, used car tires are a growing problem in Poland and worldwide. About 200,000 tonnes of used tires are coming every year in Poland. A rubber waste fire is a threat to neighboring buildings and people living nearby such a place. Thermal processing is considered as the most effective technology for tires utilization. As a result of this processing, alternative fuel can be obtained, which can be utilized by internal combustion engines. Such a process has two main advantages: the most important is waste processing, its utilization and thus the reduction of the threat to the natural environment. The second advantage is the production of fuel, which can be used for various purposes, for example fuel for vehicles or as fuel for CHP sets [1, 6, 10]. However, technologies of thermal processing including torrefaction, pyrolysis and gasification are commonly used for biomass processing focusing on biochar and syngas production, respectively.

Pyrolysis is the process of biomass decomposition at temperature of 350°C–600°C without oxygen presence. Low pyrolysis temperature is also called torrefaction (process temperature around 350°C). Under such conditions, the batch material decomposes. The following materials can be used as input material: wooden by-products from wood processing, waste from forestry, waste from the food industry (fish waste, coconut shells, corn cobs), energy crops (Virginia Mallow, *Salix Viminalis*), municipal waste (tires, selected fractions of municipal solid waste). Thermal processing of the substance results in: solid products, char (biocarbon, material with a high carbon content) and gas products. From the gas, after cooling it to ambient temperature, condensed hydrocarbon compounds. This liquid is called: bio-oil, pyrolysis oil, condensate. This fraction can be an alternative fuel used in internal combustion engines. The amount of gas and charcoal produced depends on the following: the input material, its fragmentation from the temperature and length of the process. In addition, the physicochemical properties of the liquid fuel resulting from the pyrolysis process are also varied, depending on the material

fed to the pyrolysis process [2, 4, 9–13]. In our installation for thermal processing as a result of the torrefaction of Virginia mallow, a pyrolytic oil was obtained, the composition of which has the largest share of the following compounds:

- $C_{18}H_{35}NO$ (9-Octadecenamide, (Z)-) 22.03%
- $C_2H_4O_2$ (Acetic acid) 13.54%
- $C_3H_6O_2$ (2-Propanone, 1-hydroxy-) 13.53%
- $C_5H_4O_2$ (Furfural) 11.93%
- $C_4H_8O_3$ (1,2-Ethanediol, monoacetate) 8.56%
- $C_6H_{12}O_2$ (2-Pentanone, 4-hydroxy-4-methyl-) 5.56%.

At other temperatures (300°C–375°C), the proportion of individual compounds changed. Nevertheless, these compounds are always the main components of the oil. Table 1 shows the percentage of gaseous, liquid and solid products from pyrolysis in exemplary installations described at papers [7–10].

Table 1. Comparison between individual fractions from pyrolysis process

Pyrolysis/ Torrefaction plant	Temperature	Char	Oil	Gas
	[°C]	[%]		
CSN ISO 647 [7]	520	38.5	44.5	16.8
Pilot plant [7]	600	39.2	51.8	7.9
Barbooti [3]	460	34.1	53.0	12.9
Kaminsky [8]	600	40	51	9.1
Williams [7]	600	35	55	10
Boxioing [5]	500	37.6	45.7	16.5
Our plant	350	56.3	15.0	28.7

As a result of pyrolysis/torrefaction of wastes from the wood industry or crops of *Virginia Mallow*, *Salix viminalis*, products that contain water in its structure, water is also transferred to condensate, which for this reason cannot be mixed with petroleum based fuels such as diesel fuel or gasoline. Such an alternative fuel can be used as an additive only to alcohol fuels. It is different in the case of the pyrolysis of tires that do not contain water in their structure and the condensate in the pyrolysis of rubber waste can be blended with diesel fuel and gasoline. This is shown in Figure 1, on the left there is a mixture of oil from pyrolysis of tires and diesel fuel

and oil from pyrolysis of tires with gasoline, on the right is a heterogeneous mixture of oil from torrefaction of *Virginia Mallow* mixed with gasoline and diesel. The volume ratio is 80% of traditional fuel and 20% of the pyrolysis oil or oil from *Virginia Mallow* torrefaction [6].

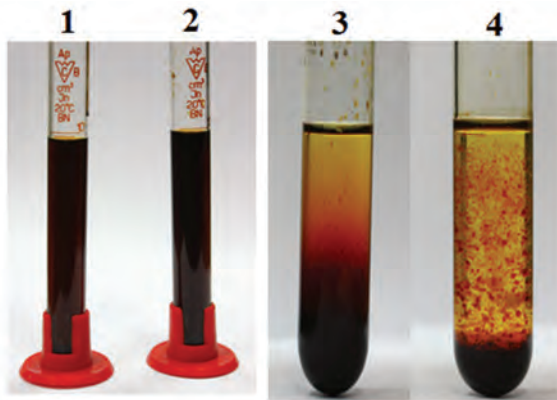


Fig. 1. Solubility oil of tyres pyrolysis in: 1 – Diesel, 2 – Gasoline, and torrefaction *Virginia Mallow* Oil blended with: 3 – Gasoline, 4 – Diesel

2. Description of the research stand

The test stand contains a 1CA90 compression-ignition piston engine by Andoria factory. Specifications of this engine are presented in Table 2. The startup of the internal combustion engine was carried out by means of a DC electric motor. The DC motor is connected in the Leonardo system with an asynchronous motor and a constant voltage generator. The rotational speed of this electric drive can be adjusted using the potentiometer. After starting the combustion engine, the DC motor was used as a brake for the engine. The in-cylinder pressure measurement was performed using a piezoelectric pressure sensor by Kistler. In addition, the engine is equipped with an encoder to measure the current

position of the crankshaft. For data recording in real time, the SAWIR our own code was used. The measurement of toxic exhaust emissions was carried out using the AI6900 flue gas analyzer.

Figure 2 shows the diagram of the test stand with the designation of individual elements.

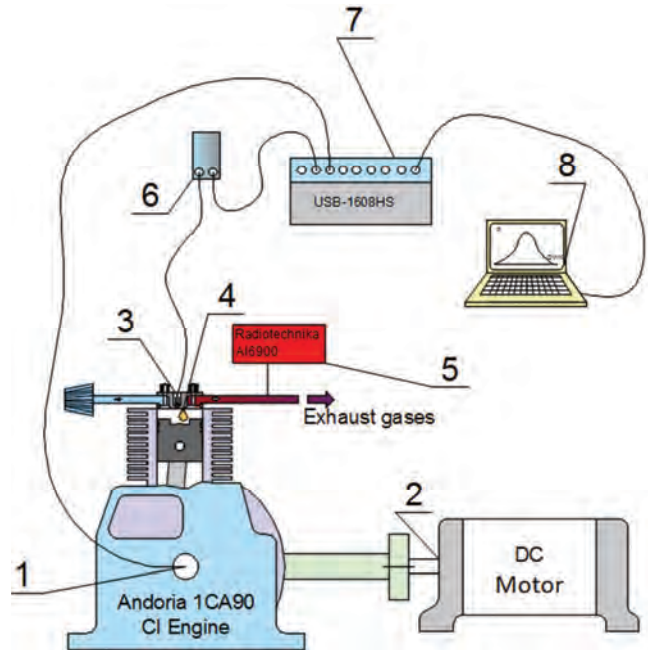


Fig. 2. Diagram of the test stand: 1 – Encoder; 2 – Ward Leonard control speed system (Load/Drive CI engine); 3 – Piezoelectric Pressure sensor Kistler 6061; 4 – Injector: Diesel/Pyrolysis Oil; 5 – Exhaust gas analyzer (CO_2 , CO, NO_x , HC, λ); 6 – Charge Amplifier Kistler 5011B; 7 – Data acquisition Measurement Computing USB-1608HS; 8 – PC with software SAWIR

Table 2. Data of the 1CA90 Andoria piston engine

Type of engine	Four-stroke
Fuel	Diesel
Number of cylinders	Single
Direct of cylinder	Vertical
Type of valve train	OHV
Number of valves	2
Cylinder bore	90 mm
Piston stroke	90 mm
Engine displacement	573 cm^3
Compression ratio	16.8
Nominal power	6.6 kW
Maximum torque	26 Nm
Specific fuel consumption at rated power output	300 g/kWh
Start of fuel injection	28° before TDC
Injection pressure	19.6–21.1 MPa
Injector nozzle	four holes
Intake valve timing	Opening: 18° before TDC Closing: 52° after BDC
Exhaust valve timing	Opening: 52° before BDC Closing: 18° after TDC
Cooling	Air cooled
Engine speed	1500 rpm

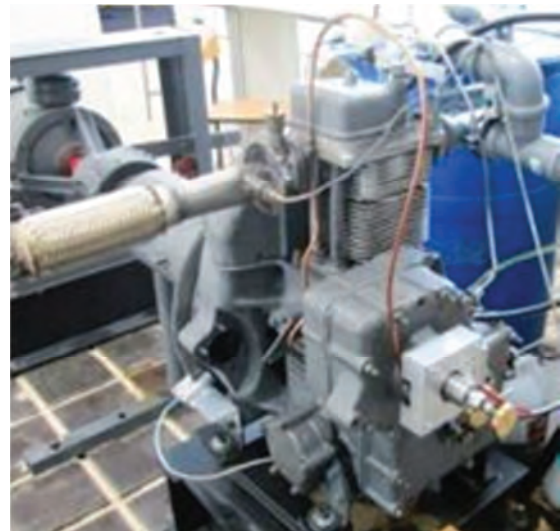


Fig. 3. Photography of the test stand

3. Analysis of results

As part of the tests, the pressure changes were recorded in the engine cylinder supplied with the reference fuel: diesel fuel (D) and diesel fuel mixtures with addition of pyrolysis oil (PO). Mixtures were created with the following shares by volume:

Table 3. Percentage share by volume of mixed fuels

Diesel	Pyrolysis oil
100% – reference fuel	0%
95%	5%
90%	10%
85%	15%
80%	20%
0%	100%

The results of pressure changes for all fuels are presented in Figure 4. The following charts present only selected fuel mixtures.

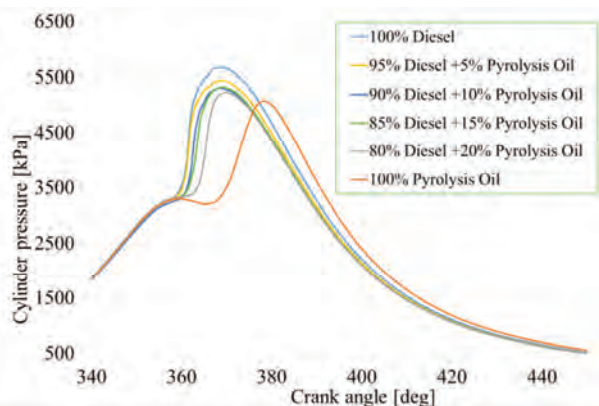


Fig. 4. Cylinder pressure vs. crank angle range 340–450°

From the presented Figure 4 it can be seen that with the increase of pyrolysis oil, the maximum pressure in the cylinder was slightly lower and the ignition was delayed. This is most evident for the pressure course when 100% pyrolysis oil is burned.

The pressure course can also be presented as a function of cylinder volume as shown in Fig. 5. You can easily identify the ignition delay when burning pyrolysis oil. Such a course suggests that injection of this fuel should be advanced.

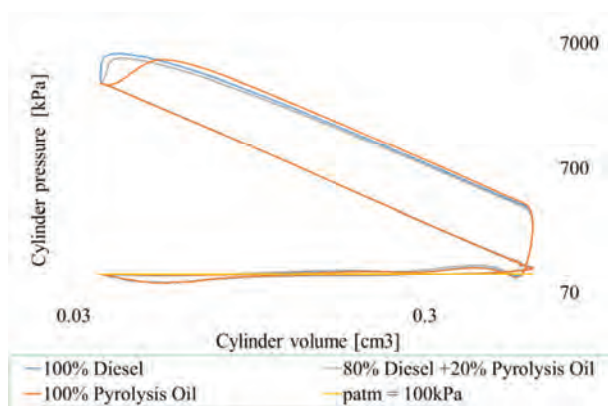


Fig. 5. Cylinder pressure vs. cylinder volume in log scale

Based on the registered pressure changes in the engine cylinder, the rate of in-cylinder pressure change was calculated and shown in Figure 6. Addition of oil from pyrolysis reduces the rate of combustion. Combustion lasts longer and is not as violent as in the case of reference fuel (diesel fuel).

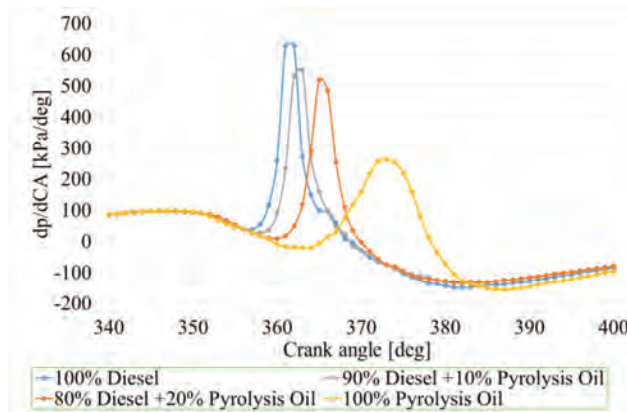


Fig. 6. Pressure rise rate vs. crank angle

One of the criteria for assessing the correct operation of an internal combustion engine is the unrepeatability of its work cycles. The indicated work was selected as the parameter determining the unrepeatability of the engine operation. In order to determine the unrepeatability of the engine operation, the following relationship was used:

$$\text{COV}_{Li} = \frac{\sigma(Li)}{\bar{Li}} = \frac{\sqrt{\frac{1}{k} \sum_{i=1}^k (Li_i - \bar{Li})^2}}{\bar{Li}} \cdot 100\% \quad (1)$$

where: $\sigma(Li)$ – standard deviation of the indicated work, $\bar{Li}$ – average value of indicate work from 100 cycles, Li_i – indicated work of the i -th cycle, k – number of analyzed cycles.

For the tested fuel, the same pyrolysis oil, the unrepeatability of the engine operation was 3.1%. This is a value not much higher than calculated for the diesel engine. For a reference fuel, the unrepeatability value was 2.6%.

The next Figure, Fig. 7, shows the value of the maximum pressure and angle of rotation of the crankshaft at which the maximum pressure occurred. Presented is a set of results for 3 fuels: reference fuel, i.e. diesel, only oil from pyrolysis, and a mixture of diesel and pyrolysis oil (80% diesel + 20% pyrolysis oil).

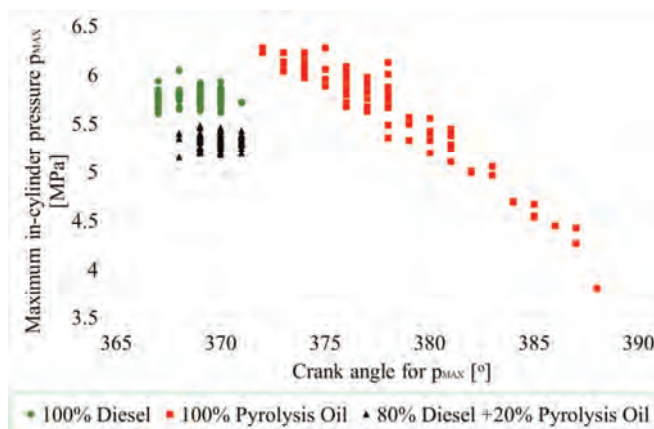


Fig. 7. Graph showing values of maximum pressure and angle of rotation of the crankshaft for which maximum pressure was recorded for 100 engine cycles

From the graph it can be seen that for the reference fuel, the maximum pressure does not have large fluctuations and

is always in the range of 367-371 degrees of crankshaft rotation. Addition of 20% oil from pyrolysis to fuel caused a slight delay of the pressure peak, in addition the maximum value decreased as compared to the reference fuel. In the case of engine operation on the pyrolysis oil itself, the maximum pressure has a large dispersion: from almost 4 MPa to over 6 MPa. In addition, the maximum pressure value occurs in a wide range of crankshaft angle (from 372 degrees of crankshaft rotation to 388 degrees of crankshaft rotation). This is a confirmation of the result of the increased unrepeatability of the engine's operation.

During the tests, emissions of toxic exhaust components were registered. The graph (Fig. 8) presents the changes in carbon monoxide emissions for various fuels from pyrolysis in fuel.

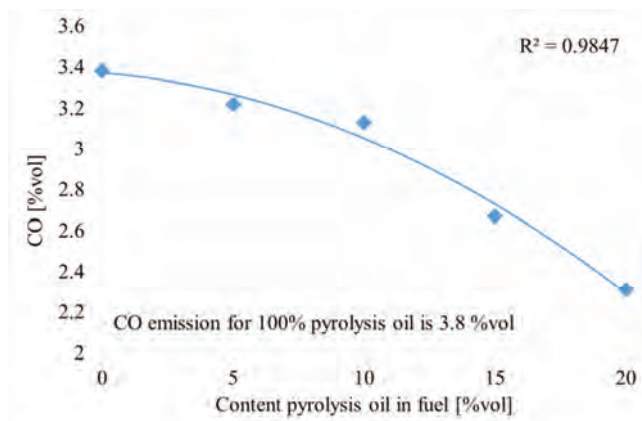


Fig. 8. CO emission versus content of pyrolysis oil in analyzed fuel

The graph shows a downward trend in CO emissions with an increase in the oil from pyrolysis in the range of 0-20%. However, the emission of CO for fuels consisting only of pyrolysis oil is higher (3.8%) than for the reference fuel (CO emission has a value of 3.4%).

The next chart shows the emission of NO_x in the exhaust (Fig. 9). In this case, a decrease in the emission of nitrogen oxides with the increase of oil from pyrolysis in the fuel was also observed. This may be due to a milder combustion process, during which temperatures are not reached as high as in the case of diesel engine operation. For an engine running only on pyrolysis oil, the NO_x emission is 259 ppm.

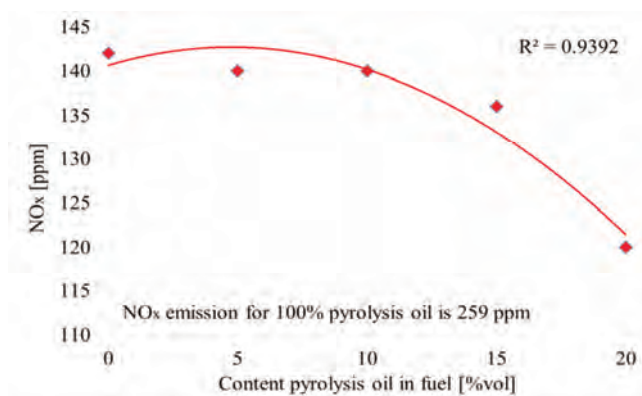


Fig. 9. NO_x emission versus content of pyrolysis oil in analyzed fuel

The last measured exhaust component are hydrocarbons, shown in Figure 10. In this case, it was not possible to determine an unambiguous trend, hence only measuring points were marked on the graph. The test was repeated twice, hence the data is the averaged data from 3 measurements. Having this on mind, the HC emission for 10% pyrolysis oil seems to be credible. The HC emission for pure pyrolysis oil was 181 ppm and was similar to other tests except 10% pyros oil.

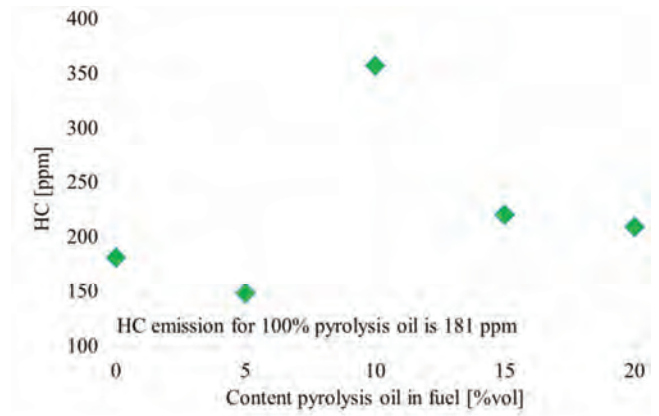


Fig. 10. HC emission versus content of pyrolysis oil in analyzed fuel

During the research, an unfavorable phenomenon of precipitation in the mixture was observed. The precipitate formed after mixing diesel oil and pyrolysis oil. The sediment on the bottom appeared after several minutes from mixing (Fig. 11). No more was formed after filtering the sediment mixture. Before mixing, the two liquids were filtered and did not contain impurities. The precipitate is a jelly-like substance that unfortunately causes the fuel filter built into the fuel system to become clogged.

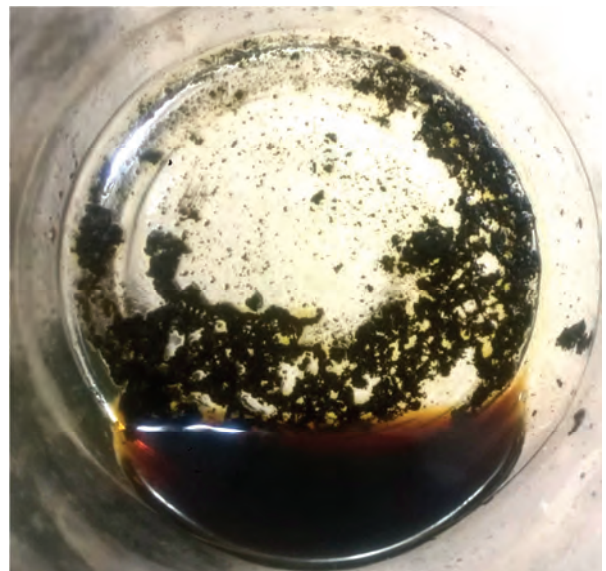


Fig. 11. Sludge from a mixture of diesel and oil from tire pyrolysis

4. Conclusion

Pyrolysis oil can be formed from the thermal processing of various substances: wood industry waste, special crops, waste from the food industry, used car tires, etc. In the

world and in Poland, there are many installations that work and offer products from the pyrolysis process. Pyrolysis reactors have different constructions and operate at different temperatures. A change in the temperature of the pyrolysis process results in a change in the share of individual product fractions produced during pyrolysis.

The pyrolysis oil has different properties depending on the batch material that is used in the thermal processing. The pyrolysis oil, which is produced from substances of vegetable origin, contains water. This component prevents the mixing of such oil with conventional fuels: diesel oil and gasoline. There is almost immediate stratification after mixing these two substances. This is the case for the gasoline / diesel fuel mixture with the Virginia Mallow pyrolysis oil. It is different in the case of pyrolysis of rubber waste. The pyrolysis oil does not contain water and mixes well with gasoline and diesel.

After a while, after mixing oil from pyrolysis and diesel, a gelatinous substance precipitates at the bottom of the container, therefore the mixture must be filtered before being fed to the fuel system. Otherwise, the fuel filter will clog and the engine will stop working. This is a troublesome problem, hence an attempt was made to start the engine only on pyrolysis oil. The trial was positive. The engine was able to run on pyrolysis oil and was constantly working during measurements.

Based on the recorded data, it can be seen that the engine working on the pyrolysis oil itself has a more unrepeatable operation. The maximum pressure has a large dispersion. In addition, the maximum pressure occurs for a wide range of crank angle rotation. This may be due to the precipitation of a jelly-like substance in the fuel system. The advantage of the engine's work on pyrolysis oil is the higher average value of the indicated work. In addition, combustion occurs less rapidly.

In order to improve the combustion efficiency of engine pyrolysis oil, it would be necessary to inject fuel earlier than conventional fuel, since the pyrolysis oil has a delayed

ignition. On average, the maximum pressure for diesel oil is for the crankshaft position 9 degrees after TDC, and for pyrolysis oil the medium of maximum pressure is 18 degrees after TDC.

The addition of pyrolysis oil to the reference fuel causes reduces the toxic components of the exhaust gas. For CO, the decrease is 31.5% with the addition of 20% of the volume of pyrolysis oil to the reference fuel. In the case of NO_x, the decrease is 15.5% with a share of 20% by volume of pyrolysis oil. Hydrocarbons has similar values for the reference fuel and mixture. In the case of engine operation on the pyrolysis oil alone, the emissions is higher than for diesel.

The injection timing was not changed in the tests presented, because it was the first step in investigation focusing on combustion progress comparison where several different fuels were applied. Therefore, injection timing optimization was not done in these tests. It is expected that toxic CO and NO_x emissions are changed with changing fuel's LHV. But, the emissions results were used only to confirm impact of combustion duration (based on HRR plots) on maximum combustion pressure-temperature, so CO and NO_x emissions, respectively.

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Nomenclature

BDC	bottom dead center	HC	hydrocarbons
CA	crank angle	IC	internal combustion
CHP	combined heat and power	IMEP	indicated mean effective pressure
CI	compression ignition	Li	indicate work
CO	carbon monoxide	NO _x	nitrogen oxides
COV	coefficient of variation	OHV	overhead valves
D	diesel	PO	pyrolysis oil
DC	direct current	TDC	top dead center
DI	direct injection		

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Experimental verification of design calculations of the internal combustion engine cooling system

The article presents a comparative analysis of calculation results and measurement results of cooling module built on the basis of design calculations. The computations were made in software dedicated to the calculations of heat exchangers. The measurements were performed on a cooling system mounted in a special purpose vehicle together with a combustion engine, which was loaded by an engine dynamometer. Based on the comparison between input data for calculations and results of measurements, differences were found, therefore changes were proposed to be made in the model.

Key words: cooling system, combustion engine, designing, numerical calculation

1. Introduction

Cooling systems of modern engines are becoming increasingly complex. This results from the high technical development of combustion engines associated with the need for thermal stabilization of many elements of both the engine and its instrumentation [1, 2]. Hence, the design of cooling systems requires knowledge of not only the principles of heat exchange, but also the construction of modern engines and the elements that require thermal stabilization. The arrangement of the elements of the cooling system and the impact of systems surrounding it is also important. Computer software with various degrees of complexity are the tool that supports the engineer in the designing process; they enable the construction of complex systems and perform highly advanced calculations. However, it is not always possible to take into account all the factors affecting the system and all the phe-

nomena taking place in it. Therefore, the ability to verify the calculation results in an experimental way is desirable. The work presents a computational model of engine cooling system, results of calculations and measurements performed on a system built using the calculation results. It also provides a comparison of the obtained values.

2. Computational model of the cooling system

The cooling system is to be applied to the engine of a special purpose vehicle. The vehicle's layout conditions require horizontal placement of heat exchangers above the powerpack. The cooling system is designed to cool the engine coolant, engine charge air, gearbox oil, hydraulic oil of fan drive and hydraulic system. In addition, the cooling system must contain a condenser for the air conditioning system of the passenger space.

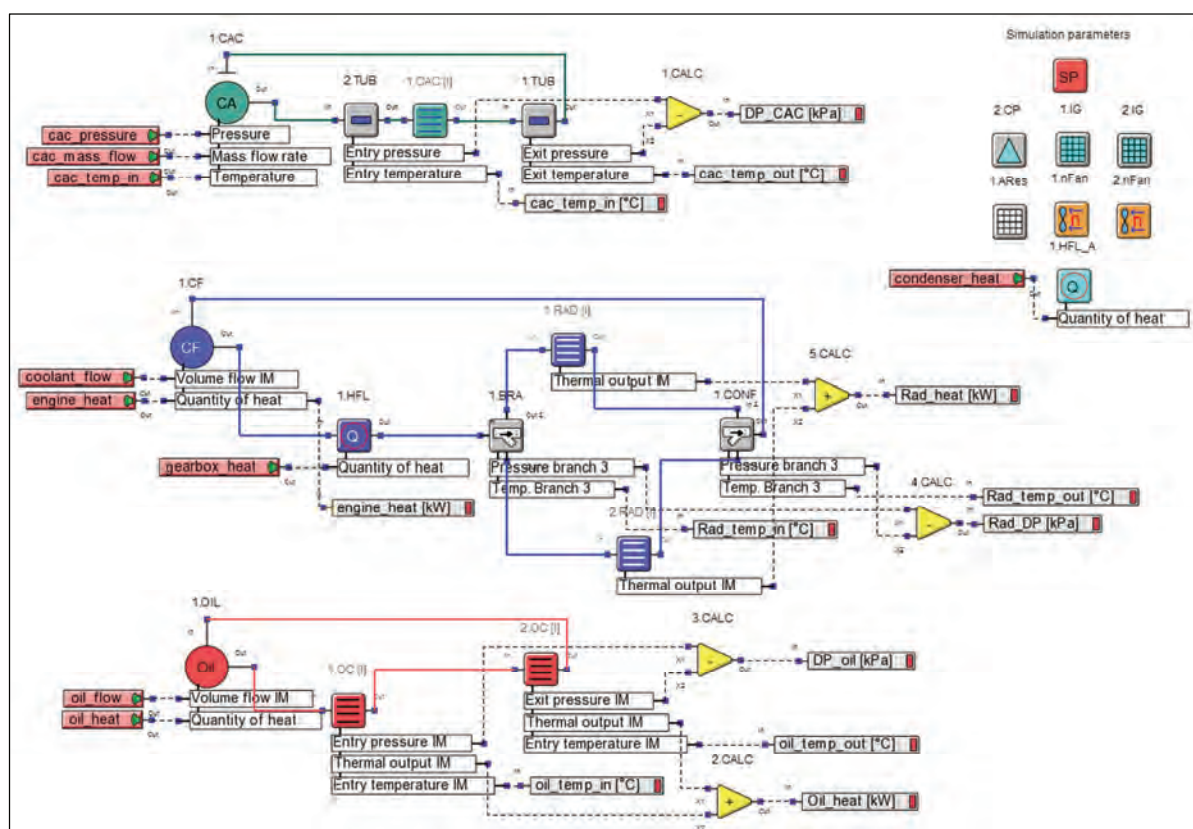


Fig. 1. 1D model of the cooling system

The calculations were performed in the KULI software, using BSPL core systems. The 1D model of the cooling system is shown in Fig. 1. The coolant circuit (blue circuit) contains heat from the engine, heat absorbed by the coolant from the oil in the gearbox exchanger and two heat exchangers in the parallel system. The heat exchanger models were built on the basis of measurements made on a heat exchanger with specific dimensions and core system. They were then scaled in the software by changing the height and length of the heat exchanger core. For a given system, the measurements were made for an insufficiently thick core. Therefore, the required thickness was simulated by parallel placement of two heat exchangers of the same size. In the cooling circuit of the charge air (green circuit), flow losses are a very important aspect, so the tube sections between the engine and the heat exchanger were taken into account. The oil cooler (red circuit) was modelled as a U-flow heat exchanger to achieve better heat exchange results.

The 3D model of the cooling system, with a preset air-flow path, is shown in Fig. 2. It shows the arrangement of the heat exchangers in the module. It is assumed that a cover in the form of blinds is to be installed in front of the heat exchangers. Due to the lack of information about the dimensions and parameters of the condenser, its influence on other elements of the cooling system was taken into account by ascribing a value of the heat absorbed from the condenser to the cooling air. This value was ascribed between the cover and the heat exchangers.

Axial fans are used to ensure forced airflow through the heat exchangers in engine cooling systems. However, the limited space for the installation of the cooling system and its special dimensions resulted in the selection of two mixed flow fans. For this reason, too, the space available for the exit air is limited to two areas with a small cross-section, on which the blinds were fitted.

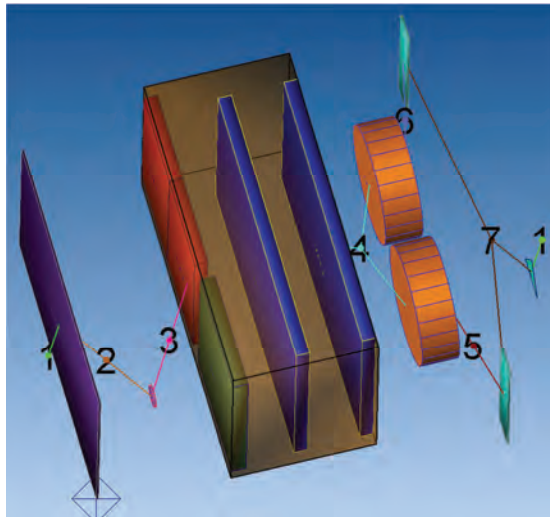


Fig. 2. 3D model of the cooling system (violet – inlet grit, green – charge air cooler, red – oil cooler, blue – radiator, orange – fan, light green – outlet grit)

3. Results of calculations

The calculations were made for an ambient temperature of 50°C because the cooling system is to be designed for

this value. The maximum temperature of the coolant at the exit of the engine can reach 105°C, the gearbox exchanger is attached to the coolant flow system after the engine but before the thermostat. 1.1 safety factor is assumed for all heat exchangers. The cooling system cover is modelled as a blind with a clearance of 50%. The air exit covers of the fans were initially modeled as blinds with a clearance of 50%, but, due to the results obtained, calculations for 70% clearance blinds were also performed. The results of the calculations for both cases, with and without heat from the condenser, are shown in Table 1.

Table 1. Results of design calculations

Value	Unit	Exit shutters 50 %		Exit shutters 70 %	
Ambient temperature	°C	50	50	50	50
Condenser heat	kW	22	0	22	0
Condenser air entry temp	°C	50	50	50	50
Condenser air exit temp	°C	54.4	50.0	53.1	50.0
CAC mass flow rate	kg/s	0.36	0.36	0.36	0.36
CAC entry temperature	°C	212	212	212	212
CAC exit temperature	°C	78.9	75.2	73.1	70.5
CAC pressure drop	kPa	5.5	5.5	5.4	5.3
CAC+ pipe pressure drop	kPa	6.1	6.1	6.0	6.0
Thermal output	kW	50.2	51.6	52.4	53.4
Oil volume flow	l/min	70	70	70	70
Oil cooler entry temp	°C	82.9	78.6	78.5	75.5
Oil cooler exit temp	°C	73.3	68.9	68.9	65.8
Oil cooler pressure drop	kPa	28.4	31.8	31.9	34.2
Oil heat rejection	kW	20	20	20	20
Coolant volume flow	l/min	240.9	240.9	240.9	240.9
Engine exit temp	°C	118.0	113.3	102.2	98.9
Transmission exit temp	°C	122.7	118.0	106.9	103.6
Radiator entry temp	°C	122.7	118.0	106.9	103.6
Radiator exit temp	°C	110.9	106.2	95.1	91.9
Radiator pressure drop	kPa	5.1	5.2	5.5	5.6
Transmission heat	kW	70	70	70	70
Engine heat	kW	105	105	105	105
Radiator heat rejection	kW	175	175	175	175
Entry shutter clearance	%	50	50	50	50
Entry shutter pressure drop	Pa	31	32	63	64
Exit shutters clearance	%	50	50	70	70
Exit shutters pressure drop	Pa	3406	3450	3027	3055
Fan 1 – Input speed	l/min	3500	3500	3500	3500
Fan 1 – Air temperature	°C	103.8	99.0	88.4	84.9
Fan 1 – Mass flow rate	kg/s	2.39	2.42	3.39	3.42
Fan 1 – Volume flow	m³/s	2.61	2.61	3.55	3.55
Fan 1 – Pressure difference	Pa	3641	3688	3462	3493
Fan 1 – Power	kW	18.7	18.9	22.8	23.0
Fan 2 – Input speed	l/min	3500	3500	3500	3500
Fan 2 – Air temperature	°C	103.8	99.0	88.2	85.1
Fan 2 – Mass flow rate	kg/s	2.39	2.42	3.38	3.41
Fan 2 – Volume flow	m³/s	2.61	2.61	3.54	3.54
Fan 2 – Pressure difference	Pa	3641	3688	3464	3496
Fan 2 – Power	kW	18.7	18.9	22.8	23.0

4. Results of measurements

The measurements were stationary, using a dynamometer connected to the shaft of the powerpack, in accordance with the procedures applied by the manufacturers of heavy-duty engines during the acceptance of engines installed in vehicles. This means that the thermostat is to be fully open at all times during the measurements, the fans are to operate at the maximum presumed speed, and the engine is to be subjected to a constant load until the temperature of the coolant is stable. Measurements are made for maximum power and maximum torque. Figures 3, 4, 5, 6 and 7 show the measurement results for the maximum engine power. During the measurements, the passenger space air conditioning system was disabled, so the condenser did not emit heat to the cooling air.

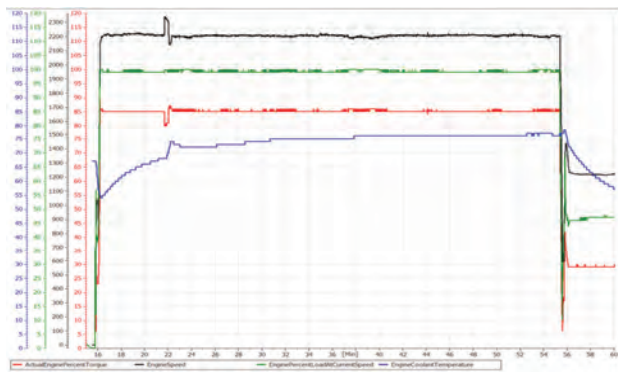


Fig. 3. Engine operating parameters during the measurement – engine speed (black), engine load (green), engine torque (red), coolant temperature (blue)

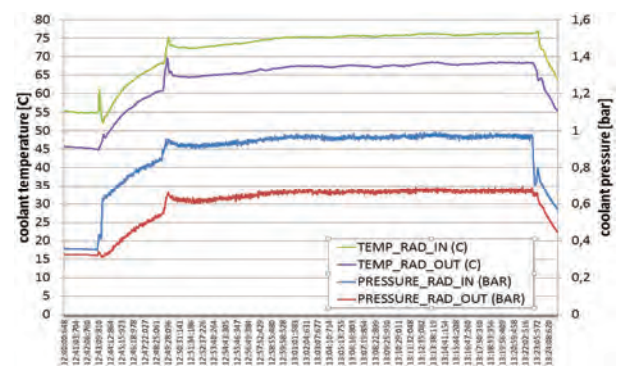


Fig. 4. Temperature and pressure of coolant measured before and after the liquid cooler

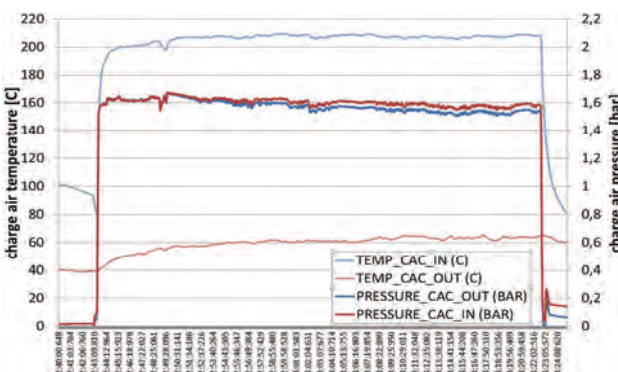


Fig. 5. Temperature and pressure of engine charge air measured before and after the air cooler

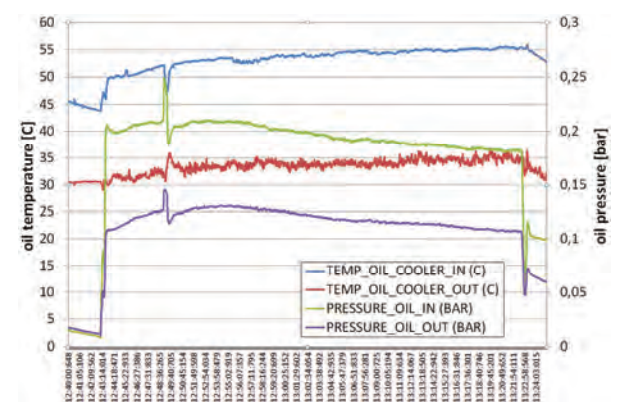


Fig. 6. Temperature and pressure of hydraulic oil measured before and after the oil cooler

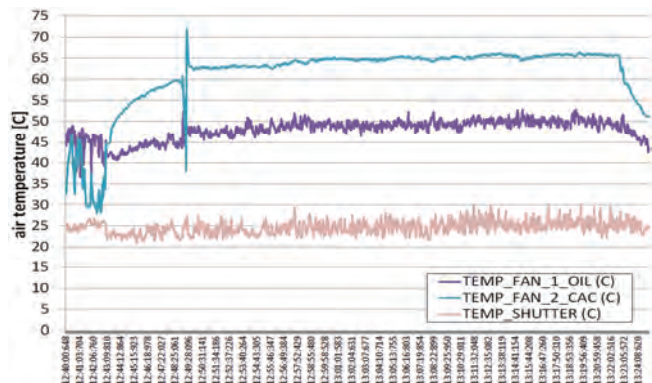


Fig. 7. Temperature of cooling air measured on the exchangers cover and on the fan cover

5. Comparative analysis of calculation and measurement results

The measurements were made at ambient temperature of 25°C. Therefore, the calculations were also performed for this value. Stabilized values of measurement results and calculation values are presented in Table 2.

Table 2. Results of measurements and calculations for 25°C ambient temperature

Parameter	Unit	Test results	Calculations	Calculations
Ambient temperature	°C	25.0	25.0	25.0
Radiator in	°C	76.0	76.8	90.4
Radiator out	°C	68.0	65.0	78.6
Radiator pressure drop	kPa	30.0	5.7	5.7
Charge air cooler in	°C	207.0	187.0	187.0
Charge air cooler out	°C	63.0	44.5	49.2
CAC pressure drop	kPa	4.5	4.8	5.0
Oil cooler in	°C	55.0	51.5	54.4
Oil cooler out	°C	35.0	41.5	44.4
Oil cooler pressure drop	kPa	7.5	39.4	39.3
Fans speed	rpm	3350	3500	3500
Air temperature out	°C	57.5	57.5	70.4
Exit shutters clearance	%	no data	70	50

The cooling system uses two fans in parallel layout, what required their separation. This was achieved by creating spiral ducts for each of the fans. Due to the applied hydraulic system of fan drive, the maximum obtained rotational speed was by 150 rpm lower than that assumed in the design calculations. The difference in oil temperature between the inlet and outlet of the oil cooler was twice as large as that obtained during the calculations.

The difference in temperature of the coolant between the inlet and outlet of the radiator during the measurements was 3.8 K lower than it resulted from the calculations. This may mean that the coolant flow was greater than that specified by the engine manufacturer to be used for calculations, as there was low flow resistance in the heat exchanger and tubes. At the same time, the heat dissipated from the gearbox to the coolant was actually lower than that provided for calculations. When the calculated heat and the temperature difference from the measurements are taken into account, the calculated volume flow rate of coolant is approximately 20 l/min greater than that given for the engine without a cooling system.

The temperature of the charge air that passed the turbo-charger was 20 K higher than specified in the engine cata-

logue card, but, as the measurements showed, the temperature of the air before the air filter was also about 17 K higher than assumed by the manufacturer. Moreover, a condenser with a front surface larger than that of the air cooler was mounted on the air cooler. This increased the resistance of the airflow through this element of the cooling system. As a consequence of the differences, the charge air temperature that passed the air cooler was about 14–18 K higher than calculated.

Table 3. Results of adjusted calculations

Value	Unit	Exit shutters 70%	
Ambient temperature	°C	25	25
Condenser heat	kW	0	0
Condenser pressure drop	Pa	238	226
CAC fouling factor	%	10	50
CAC mass flow rate	kg/s	0.36	0.36
CAC entry temperature	°C	207	207
CAC exit temperature	°C	52.2	56.3
CAC pressure drop	kPa	5.1	5.2
CAC+ pipe pressure drop	kPa	5.7	5.8
Thermal output	kW	57.1	55.6
Oil volume flow	l/min	31.8	31.8
Oil cooler entry temp	°C	55.2	55.1
Oil cooler exit temp	°C	35.3	35.2
Oil cooler pressure drop	kPa	8.1	8.1
Oil heat rejection	kW	18	18
Coolant volume flow	l/min	300.5	300.5
Engine exit temp	°C	73.1	73.2
Transmission exit temp	°C	75.8	75.9
Radiator entry temp	°C	75.8	75.9
Radiator exit temp	°C	67.4	67.6
Radiator pressure drop	kPa	11.3	11.3
Transmission heat	kW	50	50
Engine heat	kW	105	105
Radiator heat rejection	kW	155	155
Entry shutter clearance	%	50	50
Entry shutter pressure drop	Pa	63	63
Exit shutters clearance	%	70	70
Exit shutters pressure drop	Pa	3013	3001
Fan 1 Input speed	l/min	3350	3350
Fan 1 Air temperature	°C	56.9	56.8
Fan 1 Mass flow rate	kg/s	3.55	3.54
Fan 1 Volume flow	m ³ /s	3.36	3.35
Fan 1 Pressure difference	Pa	3538	3542
Fan 1 Power	kW	22.1	22.1
Fan 2 Input speed	l/min	3350	3350
Fan 2 Air temperature	°C	56.9	56.8
Fan 2 Mass flow rate	kg/s	3.55	3.54
Fan 2 Volume flow	m ³ /s	3.36	3.35
Fan 2 Pressure difference	Pa	3538	3542
Fan 2 Power	kW	22.1	22.1

Another series of calculations was made (Table 3) after introducing changes in the model that would lead to the results obtained during the measurements. The load value of the hydraulic system was verified for the lower rotational speed of the fans, what resulted in the reduction of the amount of heat dissipated to the oil cooler. In connection with the temperature change of 20 K obtained in the oil cooler, it was necessary to reduce the oil flow more than twice. The safety factor was also abandoned. The analysis of the parameters of the coolant circuit led to a reduction in the amount of heat dissipated from the gearbox exchanger to the coolant and an increase in the coolant flow. The safety factor was changed to 1.15. Since the condenser parameters were still unknown, a condenser-sized exchanger was added to the airflow before the air cooler, what increased the charge air temperature before the engine by 4 K. In addition, the charge air temperature at the inlet of the air cooler was changed to the measured temperature, resulting in a further 4 K temperature change before the engine. However, the temperature of the charge air was still about 10 K lower than during the measurements. Application of the safety factor equal to 1.5 resulted in further change in the temperature of the charge air, but it was still insufficient.

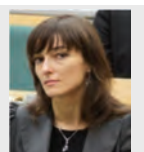
6. Conclusions

At the design stage, data is very often incomplete, so a number of assumptions must be made. In addition, as there are different suppliers of complex system components, such as the powerpack, not all parameters can be accurately estimated. Therefore, the experimental verification of the complete system parameters is a very important stage. The measurements showed differences between the results of calculations and measurements, largely due to the initial assumptions. But the layout aspects also influenced the measurement results, especially in the case of the engine charging system. This, however, can be modified. A larger number of measured parameters would allow for more accurate verification of the calculation results, including the impact of heat exchanger scaling in the software. However, based on ATB (Air to Boil Temperature), it can be concluded that the results of the calculations allowed to design a cooling system, with a condenser in its layout, that meets the requirements of operating at an ambient temperature of 50°C.

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Some aspects of long-term testing of cogeneration set fueled with biogas

Article presents the results of long-term testing of prototype of 64 kVA biogas cogeneration set (CHP – Combined Heat and Power), build with support of financial means from the MCP R&D program 1 2 1. Between others, the impact of fueling with biogas from the sewage treatment plant on the engine oil has been investigated. Used oil analyses unveiled adverse presence of silicon compounds, i.e. siloxanes, permissible level of which (according to MAN standard) was exceeded several times. In order to reduce these compounds, prototype biogas filter based on activated carbon has been developed. Based on initial calculations, overall efficiency of this cogeneration set is circa 70%. For the consumer needs, the entire installation is enclosed in a compact soundproof canopy. Besides environmental merits, the cogeneration set shows also economical advantages. If operating continuously, the gen-set CHP would produce electrical and heat power in an amount that its purchasing cost would be returned after about 12 months, under condition of having biogas.

Key words: gas engine, siloxanes, cogeneration set (CHP), bio-mass, biogas

1. Introduction

Gen-sets are independent machines generating electrical energy. Principles of their operation is based mainly on transformation of mechanical energy into electrical energy. It is done by an alternator coupled with a combustion piston engine, as a prime mover. Thermal energy created in result of fuel burning in the engine is transformed into mechanical energy in a form of crankshaft torque. Unfortunately, most of the energy from the fuel is lost with exhaust gases and into cooling system, and only 1/3 of it is transformed into mechanical work. Current combustion engine designs achieve overall efficiency not exceeding 40%, while efficiency of alternators coupled with them is in a level of 90%. The problem to increase the share of usable energy from the fuel transferred to the receiver (alternator) is an object of continuous improvements of combustion engines from the beginning of their history. From the technical point of view, small overall dimensions, compactness, portability and quick access to power source, substantially affected their popularity. Gen-sets with internal combustion engines have been used as stand-by power sources for hospitals and industry, as well as power sources in location with no power grid. This way, the history of technological progress, continuous improvements, innovations, discoveries and inventions, combined two branches of development: combustion engine and alternating current.

The principle of operation of the modern gen-sets is same as 100 years ago. During that time period, combustion engines underwent countless modernizations, in order to improve their efficiency and outputs. Nowadays, the necessity to protect environment as well as European Union's regulations on renewable energy sources, not only force but also financially support the use of alternative fuels [4]. Especially gaseous fuels coming from biomass – biogases are subject of interest. Tracking the current alternative fuel market, great interest in such fuels can be noticed within last few years, as never before. Attention is paid even to growing special varieties of plants (e.g. corn), producing new hydrolytic enzymes and highly efficient methane bacteria [5]. All these in order to improve efficiency and quality of the biogas.

Although the concept to recover thermal energy in generating sets is not a new idea and it was practiced as soon as at the end of XIX-th century, nowadays there is great interest in such type of cogeneration sets especially supplied with fuels from renewable sources. It is caused by introduction of EU directives and national laws related with energy policy. Particular attention is paid to support also small cogenerations (several dozen kW) and micro-cogenerations (up to 5 kW). The world biggest cogeneration economies are Denmark, Holland and Finland. It is worthy to underline that 82% of energy produced in Finland in 2012 came from cogeneration [8].

New regulations (Official Journal of 2018, item 1276) came into force in Poland in 2018, which promote installations producing electrical energy only from: 1) agriculture biogas, 2) biogas from landfills, 3) biogas from sewage treatment plants, 4) biogas others then listed in points 1–3 [3].

Rich literature on the use of cogeneration (especially from the last 10 years) confirms current trends at the energy market. It is obviously impossible to describe all the issues raised in the publications, but there are some aspects frequently touched by the authors. Attention is focused starting from the correct and optimal choice of power of the CHP set for given object requirements [9–12], up to calculations and economic analyses of the investments done [13–17]. Generally, looking at the cogeneration issues from the CHP sets operation modes point of view (stand-by operation and continuous operation), someone could tell these systems are easy to control and maintain. Review of the literature and own experiences show that such statements is wrong. Although I found no report where cogeneration sets made economic losses and their purchase was unprofitable, however to do so, the investor must obey the maintenance schedule and earnestly perform preventive maintenances (oil, spark plugs change, etc.). This in turn allows to avoid high repair costs and shut downs. Only this way high rate of the set operation time (availability) could be achieved, what also lead to economic profits. The literature mentioned above describes a set, for which its availability within 3-year operation period has been calculated at circa 95%. It proves very intensive and failure free operation. At such performance, investments could return as soon as within 2–

3 years. A separate issue, but related with cogeneration, is biogas production process and care to fuel the set with biogas free from harmful components. This is also fundamental for correct and reliable CHP sets operation [1, 18].

Considering the appearing market demand and current environmental trends, Sumera Motor developed cogeneration set (CHP) of 64 kVA fueled with biogas. Apart from electrical energy, the CHP provides also thermal energy recovered from cooling and exhaust systems. Water is a carrier of this energy. The CHP control system enables both parallel operation with mains as well as island operation. As of now this CHP is installed in the sewage treatment plant in Andrychów and operates continuously for about 5500 hours. Electrical energy is used for the plant current needs, and thermal energy for heating the rooms during winter time and to maintain constant temperature of one of the settlers.

Ensuring proper biogas parameters is essential for stable operation of the engine and the entire CHP. According to the engine manufacturer methane content in biogas should be minimum 55%. It should be desulfurized and dehydrated. Also pressure and flow capacity of the gas system is essential. Moreover, biogas from sewage treatment plants consists silicon compounds, so called siloxanes. They tend to settle on combustion chamber surfaces and penetrate into the engine oil, grinding off all the sliding surfaces, like bearing shells, what leads to accelerated engine parts wear out. Therefore, biogas needs to be cleaned from silicone compounds.

This article presents some aspects of long-term operation of CHP fueled with biogas from the sewage treatment plant. Between others, it presents the own-design biogas filter and its influence on siloxane presence in engine oil.

2. Cogeneration set CHP construction and general assumptions

Cogeneration set (CHP) of 64 kVA has been developed within the project RPMP.01.02.01-IP.01-12-038/16 („R&D works related with development of a new cogeneration gen-sets fueled with biogas of 25 kVA, 32 kVA, 64 kVA planned to be implemented by Sumera Motor”). This gen-set is based on Lovol's 1006-G6TAG13 gas engine and Linz Electric's PRO22S B/4 alternator (parameters presented in Table 1 and 2).

Table 1. Lovol 1006-G6TAG13 parameters

Parameters	Unit	Value
No. of cylinders	–	6
Rated power	kWe	90
Speed	rpm	1500
Frequency	Hz	50
Thermal efficiency	%	33
Natural gas consumption at 100% load	Nm ³ /h	≤ 23.8
Max. gas pressure	kPa	~90
Engine oil capacity	l	16.1

Table 2. Linz Electric PRO22S B/4 parameters

Parameters	Unit	Value
Nominal power	kVA	75
Efficiency	%	90.3
Air volume	m ³ /min	18

Apart from receiving electrical energy, thermal energy is also being recovered. It means the thermal energy that in conventional gen-set is lost via cooling system and exhaust system, here is being utilized. For this purpose, heat recovery installation has been made based on heat exchangers. Coolant/water and exhaust gases/water heat exchangers of suitable powers have been adopted. Water, which takes over the heat, flows through coolant/water heat exchanger and after initial heating up is directed to exhaust gases/water heat exchanger, where it is finally heated up. This hot water can be directly used by the user or used as next thermal energy carrier and directed to heat exchanger e.g. JAD type. Basic assumption when designing this system was to take out as much of heat from the coolant that the coolant/water heat exchanger would fully substitute the engine radiator. However, for practical reasons and as a protection the original engine radiator is kept in its place. In case of sudden coolant temperature increase above the max. allowable limit, temperature sensor sends signal to 3-way valve to open it and direct coolant to the radiator. When the coolant temperature drops down, 3-way valve closes and the heat exchange process comes back to previous circuit. Figure 1 presents the principles of operation of the heat recovery system.

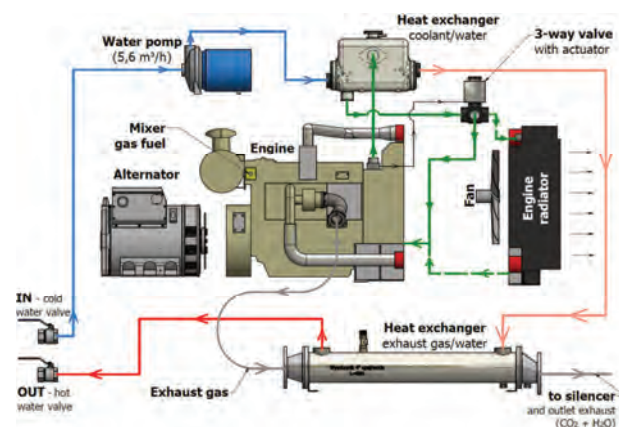


Fig. 1. Operation of the heat recovery system

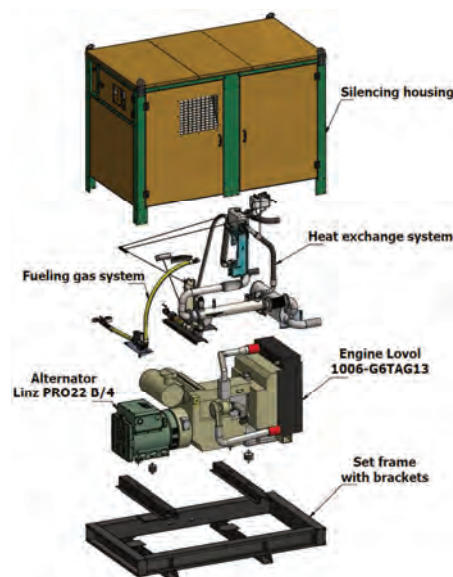


Fig. 2. Construction of the CHP 64 [kVA]

For the consumer needs, the entire installation is enclosed in a compact soundproof canopy. Figures 2 and 3 present the gen-set construction and actual photo.



Fig. 3. Side-view photo of the cogeneration set CHP 64 kVA

3. General assumptions and calculation of efficiency of the entire system

To have general view on the energy balance, theoretical calculations of total efficiency have been done. Starting point was the methane content in biogas from the sewage treatment plant and approximate energy balance for the engine used. With the NANOSENS instrument DP-27 BIO we determined 60% methane and 34% carbon dioxide content. Based on the information from the engine manufacturer, literature and our own experience, we applied the energy balance for combustion engine – as in Figure 4. This diagram shows the thermal energy used (lost) for engine cooling is equal the thermal energy converted by the engine into effective work. Theoretical calculations gave the total CHP efficiency in a level of 76%.

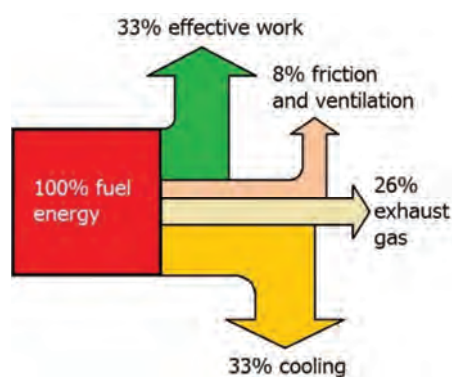


Fig. 4. Energy balance for biogas engine

For comparison, efficiency has been also determined in real operating conditions. For this purpose, the CHP has been equipped with the necessary sensors, heat meter and gas flow meter. The CHP has been loaded with 50 kW, and within half an hour heat, water inlet and outlet temperatures and gas consumption have been measured. Figure 5 shows control panel during measurements. Table 3 shows parameters measured and results of calculations of the actual efficiency of the entire system.



Fig. 5. Control panel during parameters reading. In the center, the main ComAp controller, below current water temperature readings at the heat exchangers. At the right – remote control module.

Table 3. Basic parameters related with gen-set efficiency under load of 50 kW

Parameter	Unit	Value
Calorific value of NG type E, W (according to PGNiG [6])	MJ/m ³	39.5
Biogas calorific value, W _g = 0.6 · W	MJ/m ³	24
Gas consumption, U _g	m ³	30.46
Used gas energy, E _{zuż g}	MJ	721.9
Gen-set power, P	kW	50.95
Electric energy, E _{n el}	MJ	183.42
Water specific heat, c _p	J/(kg·K)	4183
Water density, ρ	10 ³ kg/m ³	0.988
Water flow, l	kg/h	5498.6
Temperature increase, ΔT = T _{wyj} – T _{wej}	°C	12
Thermal energy, E _{n c}	MJ	276.01
Total energy, E = E _{n el} + E _{n c}	MJ	459.43
Measured efficiency, η	%	63.64

Calculations have been done with the following formulas. Energy of the gas consumed by the engine E_{zuż g} has been calculated from:

$$E_{zuż g} = U_g \cdot W_g \text{ [MJ]} \quad (1)$$

where: U_g – gas consumption [m³], W_g – biogas calorific value [MJ/m³].

The electric energy E_{n el}:

$$E_{n el} = \frac{P \cdot 3600}{1000} \text{ [MJ]} \quad (2)$$

where: P – power of cogeneration set [kW].

Thermal energy E_{n c} recovered thanks to heat exchangers has been calculated from:

$$E_{n c} = \frac{l \cdot (T_{wyj} - T_{wej}) \cdot c_p}{10^6} \text{ [MJ]} \quad (3)$$

where: l – water flow [kg/h], T_{wyj} – water temperature at the heat exchange system outlet [K], T_{wej} – water temperature at the inlet [K], c_p – water specific heat [J/(kg·K)].

The following formula presents the total energy (electric and thermal):

$$E = E_{n el} + E_{n c} \text{ [MJ]} \quad (4)$$

The total CHP efficiency η is:

$$\eta = \frac{100 \cdot E}{E_{zuż g}} \text{ [%]} \quad (5)$$

Total calculated efficiency was 76%, while the actual operating efficiency is 64%. This difference could be due to unprecise assumptions of various parameters for the theoretical model. Another cause could be also undervaluation of energy losses at each level of energy conversion in the calculation process. On the contrary, the actual CHP efficiency has been determined in specific conditions resulting from given time period. Input parameters of biogas, water as well as ambient conditions are variables in time and also could impact the final efficiency. Having the above in mind, we could state that applying cogeneration installation doubled the total system efficiency.

4. Engine oil testing

During the long-term CHP gen-set testing, also engine oil has been tested to determine impact of biogas fueling on oil contamination and consumption. The key issue was to estimate the optimum oil change interval. Oil samples were taken every 300 operating hours to be sent for chemical analyses at the specialist laboratory of FUCHS OIL CORPORATION in Gliwice. First sample showed adverse content of siloxanes of 59.8 ppm, it means five times the allowable level (according to MAN the max. allowable siloxane level is 10 ppm). Siloxanes are chemical compounds present mainly in biogas from sewage treatment plants that result from anaerobic decomposition of substances commonly occurring in soaps and detergents. When burning biogas with siloxanes, silicon is released, which could react with oxygen or other gas components. Sediments with silicon dioxides or silicates (Si_xO_y) precipitate in result of such reactions. This sediment makes a layer of white substance of a few millimeters, tending to gather on combustion chamber (valves, pistons, cylinder liner), leading to accelerated engine wear out [1, 2, 7]. This way, silicon passes to oil, also grinding out the bearing shells, etc. Therefore, further oil testing became purposeful. At the same time, activities to reduce siloxane content in biogas has been undertaken, making activated carbon filter (Fig. 6).

Taking into account the gas flow, min. time of contact with carbon (~4 s), carbon density 0.5 kg/l, a batch of activated carbon of circa 33 kg has been calculated. After 300 hours of gen-set operation with filter, oil sample has been tested and silicon content was reduced 4 times from 79 to 19 ppm – Fig. 7. Next samples showed silicates reduction to the allowable limit 10 ppm. To obtain an overview how often the activated carbon should be replaced, its weight has been measure after a month of operation. The weight of 1 dm³ was 551 g. One dm³ of completely used activated carbon should weigh circa 0.7 kg. Taking the above into account, it has been decided to replace the activated carbon every 4 months of operation, and next oil samples testing every 500 hours. Fig. 8 presents weighting of the activated carbon.

Besides silicon compounds, some other metals were noticed in the engine oil samples, like copper, tin, lead and iron. The highest content was noticed in first three samples, when the gen-set did not even reach 1.000 working hours. Most likely it was related with reaching out of engine. After about 3000 hours of operation, content of tin and lead was almost zero.

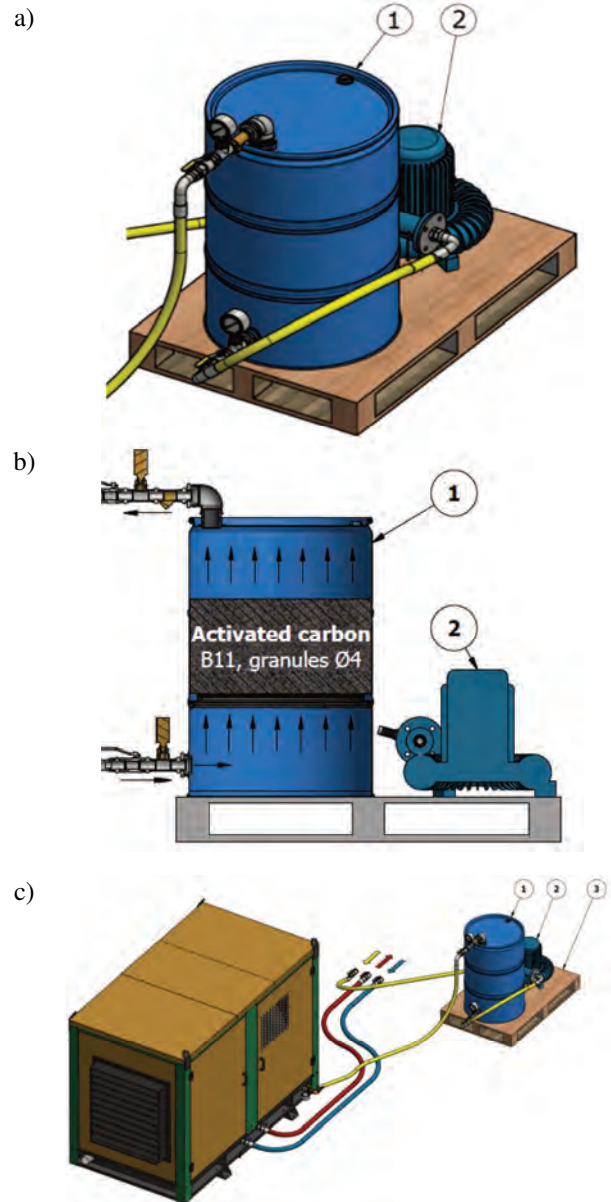


Fig. 6. Biogas filter with activated carbon, a) and b) – general view and intersection, c) view of the entire installation. 1 – filter, 2 – biogas blower, 3 – euro pallet

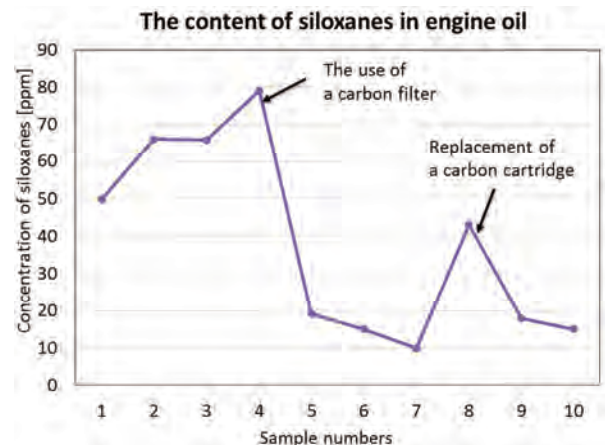


Fig. 7. Diagram presenting impact of activated carbon filter on siloxanes content in engine oil [FUCHS]



Fig. 8. Weighting of 1 l of activated carbon. Left side – clean carbon – 511 g, right side carbon after 4 months of operation – 650 g

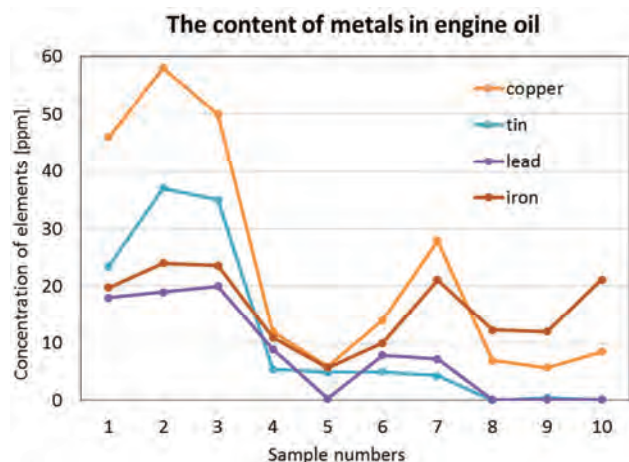


Fig. 9. Content of metals in engine oil samples [FUCHS]

Because all the other engine oil parameters, like kinematic viscosity, insoluble contamination content, soot and sulfur content and pH were for all the samples within the standard, it has been estimated the optimum oil change interval as 500 working hours (under condition of using FUCHS GANYMET ULTRA oil). It has been decided to replace the activated carbon every 2500 working hours, it means 3.5 months of continuous cogeneration set CHP operation.

5. Conclusions

The following general conclusions have been drawn from our own observations. The cogeneration set CHP described herein:

- enables continuous supply of electrical and thermal energy (with breaks for maintenance: oil, coolant change, etc.),
- thanks to heat recovery, the total efficiency of CHP compared to the engine itself is doubled,
- should operate as a stationary unit and requires continuous reception of thermal energy. It is, between others, because the engine radiator is substituted with coolant/water heat exchanger. Similarly, there is also exhaust gases/water heat exchanger. Due to high temperatures of exhaust gases, water in this exchanger should continuously circulate.
- at first start-up the water installation must be fully de-aerated,
- will undergo further improvements. It seems to be reasonable to use combustion engine without mechanical fan and substitute it with electric fan, which would turn ON only in case of possible overheating. It would improve noise emission and total gen-set efficiency by circa 4 [%].

Moreover, the use of activated carbon filter in biogas installation enabled effective reduction of silicon compounds in engine oil, what allowed to increase the oil change interval from 300 to 500 hours of operation. Attention should be drawn that after replacing the used activated carbon with a new one, the gas installation needs to be blown out with biogas, in order to remove the air. This is to avoid troubles with gen-set restarting and possible explosion.

Acknowledgements

This work was supported by the EU – project RPMP.01.02.01-IP.01-12-038/16.

Nomenclature

MCP R&D Małopolska Entrepreneurship Center

CHP

combined heat and power

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Experimental research of oil pump for a compression-ignition aircraft engine

The paper presents an analysis of the constructions of oil pumps for an aircraft compression ignition engine. It is a two-stroke liquid-cooled engine with a power of 100 kW. The system has 3 cylinders and 6 opposed pistons. The paper estimates the required oil pump capacity to make the engine components well-lubricated. Next, automotive oil pumps for diesel engines are analyzed to select a correct pump for aircraft diesel engine applications. Three pump constructions of different constructions and dimensions of a rotor were selected. A measurement bench was designed and built to test these oil pumps in the range of pump shaft speeds from 0 to 4500 rpm and volumetric flow rate up to 150 l/min. The bench also enables stabilization of oil temperature at the required level within the range from 30 to 120°C. In addition, flow resistance through engine slide bearings was simulated by changing the position of a throttling valve at the pump output to regulate pressure in the range of 0–700 kPa. The obtained capacity characteristics of individual pumps versus on oil pressure and temperature allowed us to find an appropriate oil pump to make individual engine nodes well-lubricated.

Key words: diesel engine, opposed piston engine, aircraft propulsion system, oil system, oil pump

1. Introduction

The oil pump is the key element of the internal combustion engine lubrication system. Its main function is to supply lubricating oil to various rotating and sliding parts of the engine to prevent engine wear and overheating. There are oil pumps with external and internal gearing called gerotor pumps. Gerotor pumps have epicycloidal wheels.

To select an oil pump for a specific engine, you should know the number of lubrication nodes, engine load and a type of oil system, i.e. open or closed. You should also bear in mind whether oil will flow through a cooler. The viscosity of oil and clearance are also important. Oil flow and pressure generated by the oil pump must be well matched to a given purpose. To drive an oil pump, you need power from an engine shaft, so the larger the flow and pressure in a pump, the more power is taken from an engine. Most engines do not require too high oil pressure at idle.

Speeds of rotary oil pumps can be limited by, e.g. cavitation. The authors of the paper [1] carried out some model tests of an oil pump for a motorcycle engine considering the cavitation phenomenon. The model was validated on a test stand. The tests were carried out for different pump speeds at an oil temperature of 100°C.

Internal and external oil leaks in the oil pump are further factors that can adversely affect oil flow and pressure. Certainly, the smaller clearance is, the higher pumping efficiency. Clearance, in turn, compensates for thermal expansion when a pump and oil heat up to high temperatures.

Increased oil temperature is more dependent on engine speed than load. Oil pressure, on the other hand, hardly affects the increase in lubricating oil temperature. It was observed, however, that it is fundamental to maintain a minimum pressure of oil that is supplied to a bearing because a rapid increase in a temperature of a slide bearing occurs when lubrication at low supply pressures is insufficient. Conversely, friction losses in a crank system do not depend on pressure in an oil system [5]. Pressure-controlled pumps are increasingly being used in lubrication systems.

The authors of work [4] developed a 3D model of an oil pump which enables the selection of appropriate geometrical parameters of the pump for automotive and aviation applications.

Both 1- and 3-dimensional simulations of pump were discussed in the works [2, 3, 7]. The authors demonstrated there, e.g. the use of 3-dimensional methods on gerotor oil pumps to verify 1-dimensional computations for larger parts of the governing oil system, to explain the test rig and engine experiments and to optimize the gerotor itself. Vane oil pumps and gerotor pumps were researched and these results are compared with the existing experimental data.

On the market, there are also available stands to measure pump capacity and its other parameters, e.g. ATA (Advanced Test & Automation) offers stands and tests of oil and water pumps for the automotive industry.

The first part of this paper presents the principles of a lubrication system of an aircraft opposed-piston compression ignition engine. Then, the requirements for the maximum volumetric flow rate are determined. Next, several pumps for compression ignition engines with the number of lubricating points similar to the designed aircraft engine are selected for the tests performed on a special test stand. The research allowed us to create maps of individual pumps and to select the pump for the designed engine.

2. Aircraft compression ignition engine oil pump

2.1. Engine oil system

The oil pump is designed for the new construction of aircraft engines with compression-ignition and power of 100 kW. The liquid-cooled engine has 3 cylinders and 6 opposed-pistons. The engine with a capacity of 1.5 dm³ is charged by means of a mechanical compressor.

A lubrication system in the planned aircraft diesel opposed piston engine (OPE) is a forced lubrication system with a dry oil sump (Fig. 1). The system is now equipped with two pressure pumps in a housing fixed to an engine block. The pumps are driven directly from engine crankshafts. The pump sucks oil from a tank by means of a heat exchanger and presses it through an oil filter to an oil main in the

engine block from where the oil is distributed to individual lubrication nodes. It is expected that the oil from the engine block will be pumped into the tank by high pressure generated by blows to a crankcase through pipes in the lower part of the oil sump. No additional oil pump is applied to dry the system. Such a solution can be found, among others, in Rotax 912/914 engines where a drying pump is the only element to suck oil from a charge compressor, whereas oil from the oil sump is pumped into an external tank by blow-generated high pressure in the engine block.

The theoretical flow rate of this pump is about 40 l/min for 4500 rpm, which gives the total flow rate of the both pumps, i.e. about 80 l/min. Regarding construction factors, the new type of engine will have a single oil pump with a flow rate similar to that of today's pumps to make the engine correctly lubricated and cooled.

According to the authors of the work, the [6] unit output of oil in the engine, depending on the V_{oj} slide bearings used, is from 18 to 35 dm³/(kW·h). This value is correct when the motor has one crankshaft, whereas in the case of a newly designed OPE motor this expenditure has to be doubled due to the use of two crankshafts.

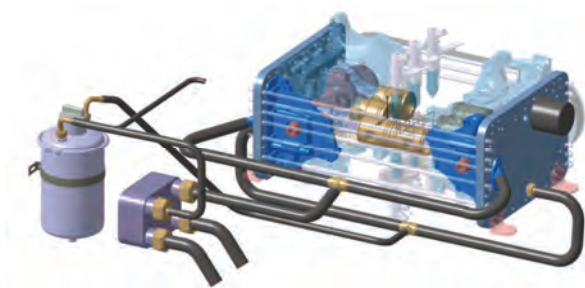


Fig. 1. External oil lubrication and cooling system OPE engine

2.2 Analysis of automotive oil pumps

The ideal pump for the new type of engine was selected after analyzing oil pumps with a similar number of lubrication points and a similar theoretical flow rate, typically mounted in the modern diesel engines. The aim was to select an automotive oil pump for an aircraft engine capable of achieving the required flow rate of 80–100 dm³/min at 4000 rpm. Such a flow rate is necessary because of two crankshafts, lubrication of the main gear and oil-sprayed piston surfaces to cool them. Three types of oil pumps of different rotor constructions and a similar flow rate were selected for our testing. Our research into flow rates will allow us to select the optimal pump construction for the OPE engine. Figures 2–4 show the pumps selected for the testing. The pumps have built-in oil pressure control valves.

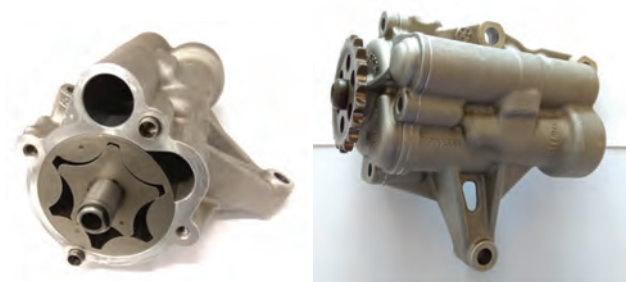


Fig. 2. Oil pump SdF, BMW M67TU D44

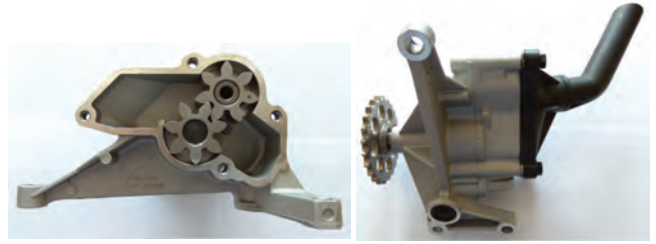


Fig. 3. Oil Pump Trucktec, Mercedes Sprinter 2.2 CDI

The Pierburg pump (Fig. 4), installed, for example, in the Nissan Patrol 2.8 TD, is a pump with variable flow rate.

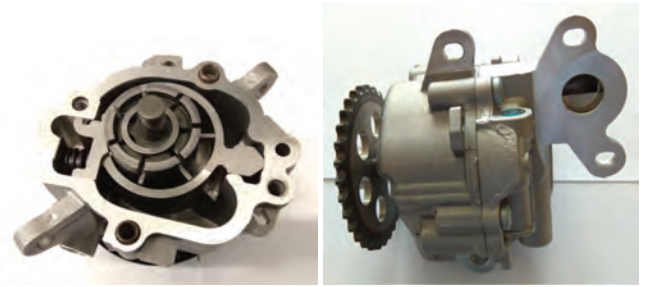


Fig. 4. Oil pump Pierburg, Nissan Patrol 2.8 TD

3. Test stand

Various types of oil pumps were tested on a special test stand where flow rates of each of the pump depending on temperature and oil pressure were measured. The pump are driven by the Tamel three-phase electric engine (4 kW) with an inverter capable of engine control to guarantee an enough power reserve to drive each of the pump and to regulate pump speeds. The pumps are driven by a gearbox with a toothed belt. The gearbox consists of two HTD M8 gears with a geometric ratio of 1:1.5 or 1:2 (depending on the pump) and a 960 mm-long and 20 mm-wide toothed belt. The engine operation is stable over the entire speed range of a maximum value of 2880 rpm.

Figure 5 shows a diagram of the oil pump test stand. The Infineon KTY 19-6M/Z temperature sensor and the Wika pressure tensiometer sensor were installed in the operating chamber of each of the oil pump. The volumetric flow rate was measured with the Badger Meter IOG 1" HF equipped with an impulser ILR740 flow meter measuring the flow rate from 5.7 to 170 dm³/min with an accuracy of 0.5%.

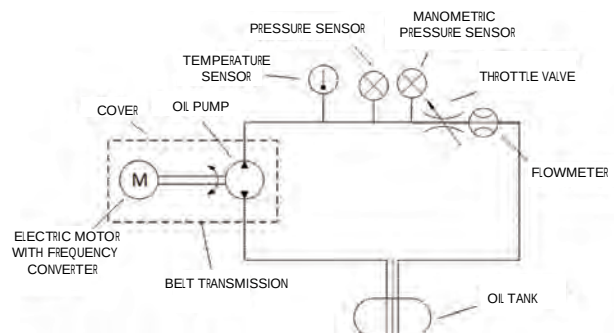


Fig. 5. Diagram of the pump test bench

The pump rotational speed was measured with a magneto-inductive sensor – Honeywell 1GT101. A 1/4 inch vol-

umetric flow control valve was mounted at the outlet from the pump chamber. Figure 6 shows the oil pump test stand.

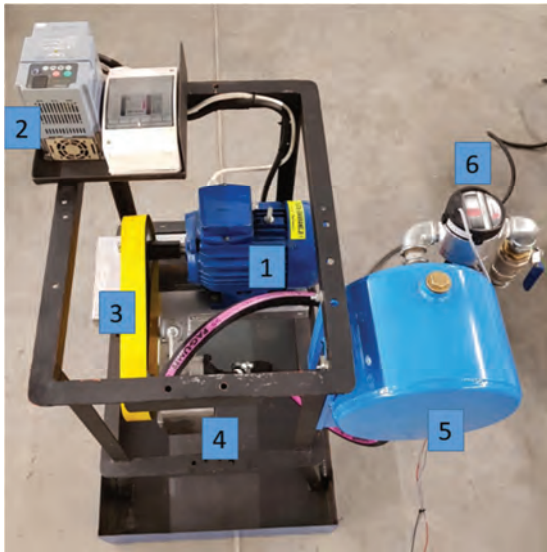


Fig. 6. Measuring station for oil pumps: 1 – electric motor, 2 – inverter, 3 – V-belt transmission with protective cover, 4 – oil pump with cover, 5 – oil tank, 6 – flowmeter

4. Test conditions

The measuring points are specified by rotational speed and the degree of flow throttling given as the pressure in the system.

The tested parameters for these measuring points were: volumetric flow rate Q_v (l/min), pressure on the pumping side p_t (kPa) and oil temperature T_{ol} ($^{\circ}\text{C}$). The engine speed was changed from 500 to 4500 rpm, every 500 rpm by means of an inverter. Throttling was controlled by changing the position of the ball valve, which meant a change of (absolute) pressure from 100 kPa to 700 kPa, every 100 kPa.

The tests were carried out at an ambient temperature of 22°C and an atmospheric pressure of 1010 hPa. The static pressure behind the p_s pump was equal to the sum of the atmospheric pressure and the hydrostatic pressure measured from the column of liquid above the sensor. The tests were performed for four oil temperature values: 25°C , 40°C , 60°C and 80°C . The oil temperature for elevated values was stabilized by means of a heating system with an electric heater and a temperature stabilization system. 5W30 oil was used for the tests.

5. Test results

The conducted tests of oil pumps used in lubrication systems of diesel engines enabled us to create flow rate characteristics for each of the pumps. Figures 7–9 show the selected results of the flow rates as a function of the rotational speeds of the pump rotor in relation to the pressure at a temperature of 80°C .

The first tested pump was the SdF pump installed, e.g. in the BMW M67TU D44. It has the highest flow rate. The maximum flow rate at a speed of 4500 rpm is about 105 l/min when the pressure at the pump outlet is equal to atmospheric pressure. The pump shows a relatively linear flow rate over the entire operating range, regardless of the

oil pressure and temperature. For an oil temperature of 80°C and a pressure of 700 kPa, the maximum flow rate is 92 l/min, which is less by approx. 13 l/min.

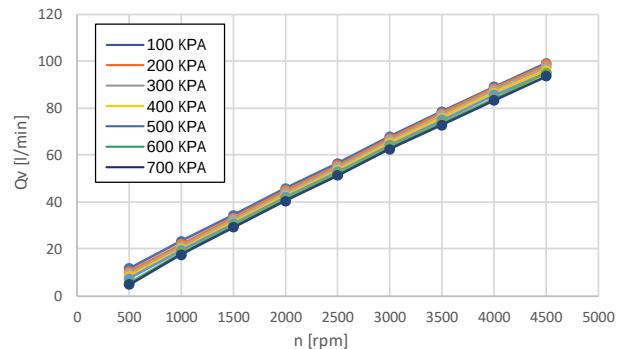


Fig. 7. Volume output as a function of speed for defined pressure values for SdF oil pump

Figure 7 shows the volumetric flow rate for the Pierburg pump as a function of speed. The measurements were without flow throttling for speeds ranging from 500 rpm to 4500 rpm. The obtained correlation of the flow rate and the pump speed is linear nearly in the entire speed range. The volumetric flow rate of the pump varies from 10 to 89 l/min.

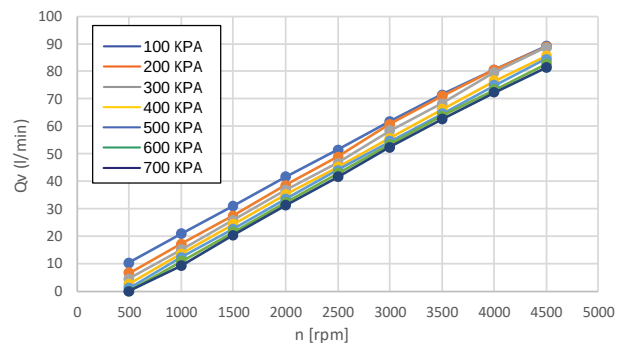


Fig. 8. Volume output as a function of speed for defined pressure values for Pierburg oil pump

The flow rate for the Trucktec pump (Fig. 8) for the pressure range of 100–500 kPa is linear over the entire speed range of the pump rotor. The maximum flow rate is 78 l/min at a speed of 4500 rpm. When the oil pressure was increased depending on the speed, the pump flow rate changed a lot.

100 KPA
200 KPA
300 KPA
400 KPA
500 KPA
600 KPA
700 KPA

Fig. 9. Volume output as a function of speed for defined pressure values for Trucktec oil pump

The pump generates a much lower flow rate at a pressure of 700 kPa, which is related with opening the bypass valve above 600 kPa.

Figures 10–12 show the dependence of the pump output on the rotational speed for different oil temperatures. In each case, an increase in the oil temperature caused a decrease in the oil pump output.

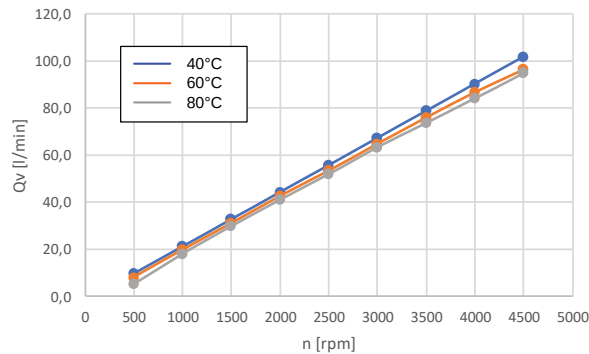


Fig. 10. Volume output as a function of speed at 600 kPa and a defined temperature value for the BMW SdF oil pump

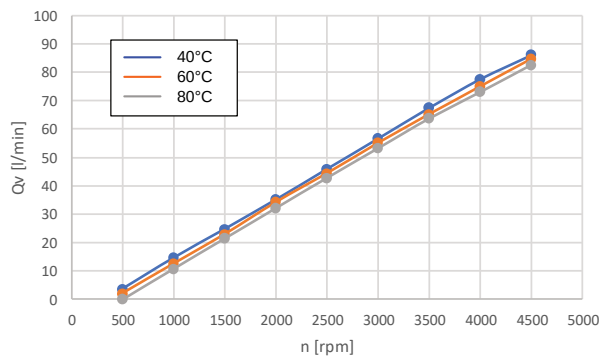


Fig. 11. Volume output as a function of speed at 600 kPa and a defined temperature value for the Pierburg oil pump

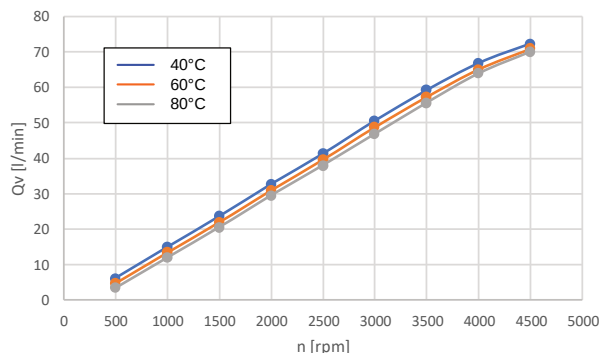


Fig. 12. Volume output as a function of speed at 600 kPa and a defined temperature value for the Trucktec oil pump

The highest decrease was observed for the SdF BMW pump, about 7 l/min, for other pumps this decrease did not exceed 3.5 l/min.

Figure 13 presents a comparison of the flow of three tested pumps for oil pressure of 600 kPa and temperature of 80 °C. The SdF BMW SdF pump had the highest flow rate of 94 l/min, while the Trucktec pump had the lowest flow rate. The difference was about 24 l/min.

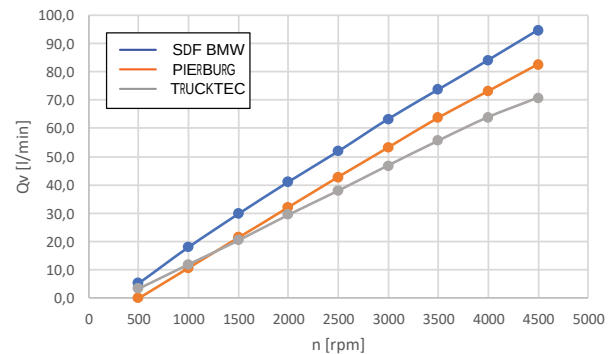


Fig. 13. Oil pump volume flow for 600 kPa pressure and 80 °C oil temperature

6. Summary

The tests of the flow rates of the automotive oil pumps were to gain certain necessary data, i.e. changes in pressure, flow rate and the impact of oil temperature on flow rate and develop an assembly of pumps efficiently lubricating the OPE engine. The obtained results enabled us to create the characteristics of the pump operations. It was noticed that the flow resistance has a significant impact on pump's operating parameters, i.e. flow and pressure. The increased oil temperature decreased the pump flow rates in each case. The maximal drop was 9% for the Trucktec pump. The obtained necessary data includes: data on the construction and operation of oil pumps, the maximum flow rates of each of the pump, operating pressure ranges, the correlation between pump flow rates and pressures and temperatures for a given pump speed, and operating ranges for oil pressure control valves.

Acknowledgement

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Nomenclature

OP Oil pump
OPE Diesel Opposite Piston Engine
RPM Revolutions per minute
ICE Internal combustion engine

CI compression ignition
VFR Volumetric flow rate

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Research and analysis of the results from the goniometer and tribotester using the diamond – derivating coating on piston ring

The elements creating friction pairs are described to be very difficult in terms of determining all optimal parameters in an unambiguous manner. The nowadays research on stability of friction pairs is focused on the surface and the top layer of surface. The main goal is find new design solutions and materials, achieving a lot of kilometers to main repair in combustion engines. The biggest structural difficulties are noticeable in friction pairs where can be observed sliding and returning motion, what is connected with sealing function. A typical example of such pair is piston ring – cylinder. Engineers work on defining an additional factor that would be able to gain the reduction of tangential force by reducing the friction coefficient in elements of friction pair. Molecular structure and nature of the bonds between the molecules presented in the material, result that the free surface energy may be such a factor, because determines the tribological properties of the material, what is reflected in the stability of the units. The work carries out of an analysis after research on the goniometer in the presence of water and diiodomethane that allowed to measure wetting drops on the surface, thanks to which can be calculated surface energy. Within this work will be performed study of the energy state of the surface of piston rings with a coating including a diamond powder. After analysis on the basis of the collected measurements will be developed conclusion regarding the potential relationship between the free surface energy and wearing of the material.

Key words: piston wearing, diamond coating, goniometer, free surface energy

1. Introduction

This article presents the research from the goniometer on the newly developed sealing piston ring for an internal combustion engine, where the novelty is an innovative – derivative coating with very good anti-wear properties with higher hardness than traditional materials (10–60 GPa), high adhesion to the substrate, high mechanical, physical and chemical stability or coefficient of friction below 0.1 [3].

Friction pairs are designed from elements, where is very difficult to define in unequivocal way all optimal parameters and research on their stability is focused on the surface and the top layer of surface. The main goal is find new solutions for design and material, thereby achieving so many kilometers of mileage to main repair in the case of internal combustion engines.

Due to the big structural difficulties that are noticeable in friction pairs where we can observe sliding and returning motion that is also connected with sealing function, engineers are currently seeking an additional factor which would be able to gain the reduction of tangential force by reducing the friction coefficient in elements of friction pair during operations.

Such a factor may be the surface free energy that results from molecular structure and nature of the bonds between the molecules presented in the material. Components of surface free energy (dispersion and polar) determine the tribological properties of the material, which is reflected in the stability of the units.

The surface free energy and its components can be obtained thanks to measurement of the wetting angle performed within the goniometer and Owens-Wendt calculation method [2].

Then, the influence of surface free energy was tested on the tribotester, checking the wearing of piston samples.

2. Research on the goniometer

2.1. Preparation

One of the possibilities to calculate the free surface energy of the element is measurement of wetting angle of the wetting drop using special instrument which is goniometer. Wetting angle can be measured with the use of the standard liquid with known value of the surface free energy and its components. This angle has a big meaning for lubrication processes and tribological point of view.

The wetting angle measurement was performed using a goniometer from the German company KRÜSS GmbH. According to the procedure, the free surface energy was determined using the Owens-Wendt method, which consists in measuring the surface contact angle using drops of standard liquids, one polar and the other one – non polar. In the research, the used pair of such liquids was water and diiodomethane. Ten partial measurements were made and after that the value of the wetting drops was averaged for each sample. During the test were performed constant conditions: the air temperature was 23°C and the humidity was at 53% level. These parameters are very important, because can have impact on the arranging drops on the tested surface (for example drop can evaporate partially and affect the results).

The measurements were performed on the sample surface that was previously cleaned with alcohol to avoid the contamination of the surface and the drop size was 2 µl (both for water and diiodomethane) to minimize the spreading of the drops under the influence of gravitational forces, that can make the wetting angle lower.

Objects of the research were piston ring samples containing in the coating the inclusion of diamond powder into the porous chrome, at the same time creating a diamond – derivating coating. For the research on the goniometer were prepared samples marked as PP, T-1, T-2, T-3, T-4 and T-5.

The sample marked as PP was a comparative sample, without diamond powder in the coating – only galvanic chromium was applied on the surface. The rest mentioned samples marked as T-1, T-2, T-3, T-4 and T-5 contained in the coating the inclusion of diamond powder into the porous chromium in various weight amounts with a granulation of 0.52 μm , as detailed in the table below (Table 1).

Table 1. The amount of diamond powder for each sample

Sample	C [%]	AL ₂ O ₃ [%]	Cr [%]
T-1	0.84	0.62	98.84
T-2	0.89	0.09	99.07
T-3	0.54	0.09	99.42
T-4	0.80	0.09	99.15
T-5	0.49	0.07	99.48

2.2. Results

The Table 2 below presents the average contact angle for each sample, calculated on the basis of the results obtained during the measurements on comparative sample and samples coated with a diamond coating using the pair of standard liquids, which were water and diiodomethane.

Table 2 Average wetting angle for samples

Sample	Average wetting angle for water [°]	Average wetting angle for diiodomethane [°]
PP	75.7	49.1
T-1	75.9	46.3
T-2	77.2	46.0
T-3	73.6	46.2
T-4	74.0	46.3
T-5	70.7	36.4

To calculate the surface free energy (γ_s) was used the Owens-Wendt method:

$$\gamma_s = \gamma_s^d + \gamma_s^p \quad (1)$$

$$\gamma_{SL} = \gamma_s + \gamma_L - 2(\gamma_s^d \gamma_L^d)^{\frac{1}{2}} - 2(\gamma_s^p \gamma_L^p)^{\frac{1}{2}} \quad (2)$$

where: the index 'd' indicates the dispersive component of the free surface energy, respectively solid material (S) or liquid (L), and the index 'p' indicates the polar component of the free surface energy, respectively solid material (S) or liquid (L).

In addition, it is known that:

$$\gamma_L(1 + \cos\Theta) = 2[(\gamma_s^d \gamma_L^d)^{\frac{1}{2}} + (\gamma_s^p \gamma_L^p)^{\frac{1}{2}}] \quad (3)$$

And after transformation the following formula can be obtained:

$$\gamma_L \frac{(1 + \cos\Theta)}{2} = (\gamma_s^d \gamma_L^d)^{\frac{1}{2}} + (\gamma_s^p \gamma_L^p)^{\frac{1}{2}} \quad (4)$$

To be able to define the value γ_s of the above equation, the measurements of wetting angles (Θ) for two different liquids γ_L^d and γ_L^p were made and the system of two equations was solved (5). Water has the following known values of surface free energy and its components: $\gamma_L = 72.8 \text{ mJ/m}^2$, $\gamma_L^d = 21.8 \text{ mJ/m}^2$ and $\gamma_L^p = 51 \text{ mJ/m}^2$, while diiodomethane is

characterized by the following values: $\gamma_L = 50.8 \text{ mJ/m}^2$, $\gamma_L^d = 48.5 \text{ mJ/m}^2$ and $\gamma_L^p = 2.3 \text{ mJ/m}^2$.

Basing on the Żenkiewicz book [4], after the transformation, the basis for determining the free surface energy value is the system of two equations:

$$\begin{cases} (\gamma_s^d)^{\frac{1}{2}} + 1.53 (\gamma_s^p)^{\frac{1}{2}} = 7.80 (1 + \cos\theta_1) \\ (\gamma_s^d)^{\frac{1}{2}} + 0.22 (\gamma_s^p)^{\frac{1}{2}} = 3.65 (1 + \cos\theta_2) \end{cases} \quad (5)$$

where: θ_1 – water wetting drop angle, and θ_2 – diiodomethane wetting drop angle.

After the calculations, were obtained the following values of dispersion and polar components of free surface energy that are presented in the following table.

Table 3. The value of free surface energy and its components

Sample	γ_s^p [mJ/m ²]	γ_s^d [mJ/m ²]	γ_s [mJ/m ²]
PP	7.92	29.38	37.30
T-1	7.25	31.13	38.38
T-2	6.51	31.63	38.14
T-3	8.53	30.62	39.15
T-4	8.32	30.66	38.98
T-5	8.37	35.42	43.79

where: γ_s – free surface energy [mJ/m²], γ_s^d – disperse component of free surface energy [mJ/m²] and γ_s^p – polar component of free surface energy [mJ/m²].

The figure below (Fig. 1) presents the chart with the components of surface free energy for each sample.

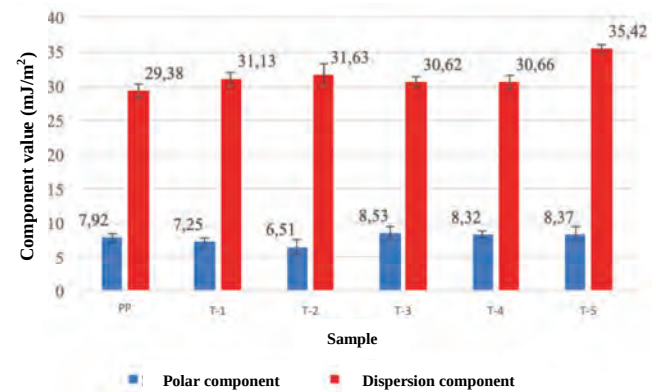


Fig. 1. Free surface energy components values for samples

The measurement results indicate high values of the dispersive component which is the derivative of dispersion interactions between molecules. At the same time, the very small value of the polar component is maintained, that tells about the uneven distribution of atoms, hence its value should be as low as possible because it affects the phenomenon of wetting and formation of the hydration layer on the surface of a solid. The total value of the surface free energy is also not large. An example to compare, for nitrided steel X90CRMOV18, which is an example of material presenting good tribological properties, the sum of dispersion components (33.01 mJ/m²) and polar (5.45 mJ/m²) gives the value of free surface energy about 38.46 mJ/m², so can be assumed that the values are similar and could indicate

good properties of a diamond derivating coating. From the presented results in the Table 3, the sample T-2 was recognized as potentially the best sample from tribological perspective due to the smallest polar component and at the same time the value of the dispersion component is relatively slightly different from the others.

2.3. Comparative research

The results obtained within the goniometer research and the influence of free surface energy, as well as its components, on tribological properties of the piston ring sample representing element of friction pair was checked on the tribotester due to using it is possible to determine the wear characteristics of material. The tested elements on the tribotester device were a sample (roll) performing a rotary motion and a counter – sample (block) that was stationary, giving 100 mm² of contact surface. The samples had the same composition of coating and were named in the same way: PP, T-1, T-2, T-3, T-4 and T-5. The friction path assigned to each sample was about 50 000 meters.

Tests on the tribotester were carried out on a continuous, close to the maximum load of the assembly that does not occur in reality. Due to the fact that the basic significance during research on the tribotester is attributed to the loads of the tested friction pairs, including unit pressure and linear speed, the linear speed was determined for 1.25 m/s (equal to approx. 350 rpm) and a load was set to 5 MPa. Thus adopted the low velocity and fairly high load values generate extremely difficult conditions in which friction pairs can cooperate and resemble those appearing in the internal combustion engine in the vicinity of ZZ of the first piston ring, at the beginning of the expansion stroke.

Table 4. Average values obtained after tribotester test

Sample	Average wearing value [μm]	Average friction coefficient value [–]	Average friction force [N]
PP	39	0.022	11
T-1	17	0.017	9
T-2	16	0.014	7
T-3	20	0.015	8
T-4	23	0.015	8
T-5	21	0.031	16

The different friction coefficients and wearing values obtained during tests on the tribotester for each sample confirm the differences in the value of free surface energy and its dispersion and polar components. Differences in surface free energy values for each sample also caused differences in the friction coefficient and wearing values of the tested material combinations. The following figures (Fig. 2 and Fig. 3) show average values of the wearing and values of free surface energy depending on the % content of C in the composition of the tested samples.

The diagram (Fig. 2) presents how the value of average sample wearing changed. The research shows that the T-2 sample was characterized by the smallest mean total consumption (16 μm), the lowest average friction force (7 N) and the lowest coefficient of friction (0.014) after 50,000 m of the friction path.

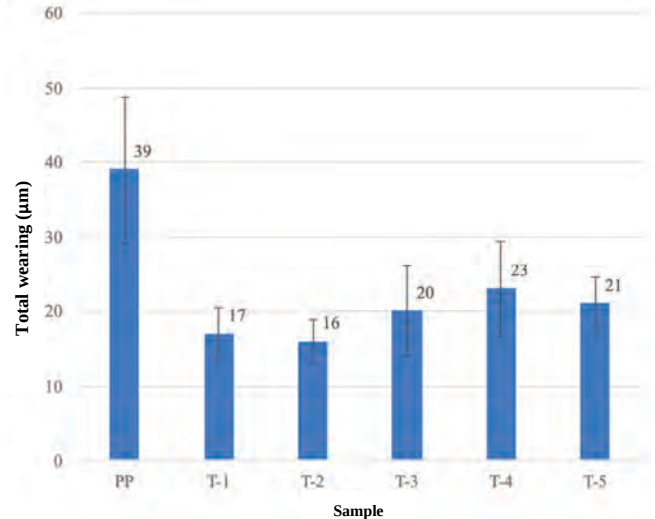


Fig. 2. Total wear of samples

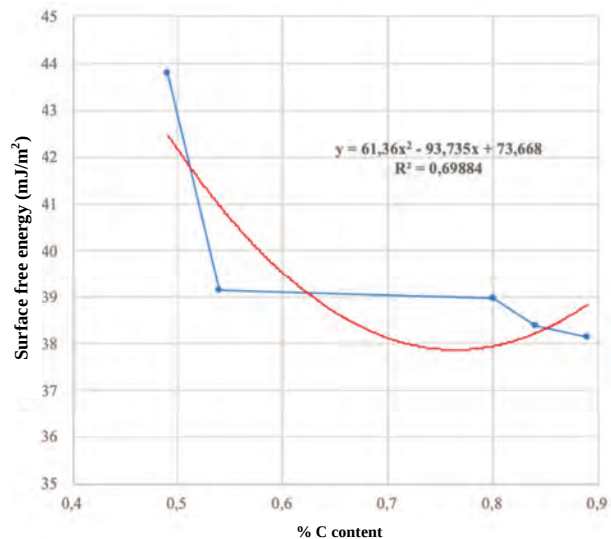


Fig. 3. Relation of free surface energy on the percentage content of C

The diagram above (Fig. 3) shows how the value of free surface energy changes with the change of the % C content in the coating composition. This relationship is roughly linear and shows that as the content of the diamond embankment increases, the value of this energy decreases.

3. Conclusion

The surface free energy determined for the samples was at a similar level to that mentioned for steel X90CRMV18, known in terms of tribological properties and successfully used in industry [2]. This could indicate justified attempts to undertake a new parameter, which is surface free energy and its polar components during the selection of materials for elements of friction pairs, the more so because it assumed characteristic values depending on the material composition of coating.

The energetic state of the surface was determined by the measured value of the wetting angle using standard liquids in the measurements and on this basis it was possible to calculate free surface energy and its components – dispersion and polar that allow getting information on interfacial interactions and surface wettability, which in the

other hand shows impact on wearing and tribological properties.

It is observed that the differences in values of the wetting angle, describing the wettability of coatings by model liquids, are reflected in the total free surface energy. Together with the increase in the content of % C in the composition of the diamond coating, polar interactions decrease.

Calculation of the free surface energy value and its dispersive and polar components have been carried out using piston ring samples of the internal combustion engine. It has shown that can affect the selection of material for friction elements. These considerations were verified during wear test performed on the tribotester.

It was also observed that the average cumulative wearing determined for a PP sample that did not have a diamond coating was the largest and was 39 μm . For the sample that had a diamond coating with the highest % content of C in the composition (0.89%), the wearing was the smallest, only 16 μm . Judging by the average values of the friction coefficient and values of tangential force, the T-2 sample was the best one from tribological point of view. At the same time, it had the smallest value of the polar component.

The obtained results show that on the basis of the surface free energy values and its components, it is possible to

select the material. As a result of the research work carried out on piston ring samples, the best results were obtained for the sample designated as T-2, including the smallest value of the polar component corresponding to the smallest wear of this sample.

The obtained results on the goniometer and tribotester indicate that it is possible to shape the surface wear of diamond – derivating coatings with the determination of free surface energy and its components.

Additionally diamond – derivative coatings are mainly characterized by a lower friction coefficient and a much greater resistance to wear in comparison to rings that are covered by the common super hard coatings. Without a doubt, the application of such coatings will have an impact not only to extend the life of system piston - ring - cylinder, but also will reduce fuel consumption even under the most strenuous conditions of work of the unit.

Solutions to the given research problem is based on the results of the goniometer test. It allows acquiring new knowledge regarding diamond coatings with a specific weight percentage composition. Positive results realized in researches pose a real chance to increase the quantities of produced rings with diamond coating, for example: sport cars, often working in high parameters.

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Researches on a photocatalytic device for toxic hydrocarbons removal from IC engine exhausts

Hydrocarbons, especially aromatic volatile organic compounds (AVOCs) and polyaromatic hydrocarbons (PAHs), emitted by IC engines, indisputably have a negative impact on the environment and public health. Even though current legislation strongly impacts on their low concentration in the exhausts, not quantity but the quality of those substances determines fumes toxicity. The paper presents results of the researches on a photocatalytic device dedicated to BTX (benzene-toluene-xylene) removal from IC engine exhausts, which can be applied in a vehicle cabin ventilation system. According to the results, the system is able to decrease toxic compounds concentration (caused by mobile emission sources) in inlet-air delivered to the vehicle interior. In effect, the system application reduces the users of a vehicle to exposure on contaminants.

Key words: photocatalysis, engine exhausts, vehicle interior, public health

1. IC engine exhausts as a vehicle interior pollutant

Unfavourable health effects of road traffic in cities, especially with large traffic volume, characterised by numerous traffic congestion, are bound in a way direct primarily with the quality of atmospheric air and acoustic climate.

Products of the combustion, such as nitrogen oxides, carbon monoxide, hydrocarbons, particles, and even metal compounds as well as biocide combustion products (substances added to the fuel in order to prevent corrosion of the tank and biodegradation of fuel) emitted into the atmosphere can be easily absorbed by the human body through the respiratory system and dermal exposure. The relation between exposure to increased pollution levels and the mentioned effects is confirmed in many researches [1–3]. The negative impact on health can have itself staying in a car interior where the level of pollutants can be several times higher than in the surrounding air [1–3].

From mentioned compounds emitted by IC engines, especially hydrocarbons (HC) are recognised as toxic, carcinogenic and mutagenic substances. Among them, aromatic hydrocarbons are the most dangerous group.

Aromatic hydrocarbons emitted by combustion engines can be divided into two, important in environmental aspects, groups [1]:

- Aromatic volatile organic compounds (AVOCs) – are organic chemicals that have a high vapour pressure at normal room temperature. Their high vapour pressure results from a low boiling point, which causes large numbers of molecules to evaporate or sublime from the liquid or solid form of the compound and enter the surrounding air, a trait known as volatility.
- Polyromantic hydrocarbons (PAHs) – are hydrocarbons containing only carbon and hydrogen, composed of multiple aromatic rings (organic rings in which the electrons are delocalized). Even though the particles are not volatile (they are relatively heavy) PAHs can be ad-

sorbed on particles, cumulate in the environment and, because of its high toxicity, cause severe problems in ecosystems.

Indisputably interior of a car cabin is an essential environment of contemporary human life. Specificity of car vehicle operation in cities, especially in road congestion conditions, effects in interior atmosphere pollution. Whereas particle matter (PM) is mostly stopped on filters applied in the vehicle ventilation system, a gaseous fraction of the exhausts can easily get into the car interior.

Despite the fact, that the problem of car cabin pollution caused by carcinogenic and mutagenic volatile organic compounds is well known, there are no general standards of its limitation. It can be assumed that a vehicle interior can be treated as an indoor space, like building [1]. In this aspect, the concentration of particularly toxic AVOCs, i.e. benzene, toluene and xylenes (BTX) concentration, very often is a few times higher than the allowable concentration of those substances for indoor space, category A (permissible for human permanent stay – Table 1) [1].

Table 1. Allowable concentration of indoor substances, category A [1]

Compound	Allowable concentration indoor (Category A) [mg/m ³]
Benzene	0.01
Toluene	0.2
Xylenes	0.1

The BTX exposure can effect in serious health problems (long-term exposition) and irritate nervous system which consequence can be a direct influence on vehicle active safety (driver distraction, lowering driver cognitive abilities etc.). This fact should focus the attention of scientists and designers on the problem of effective car cabin inlet air purification and decreasing of gaseous toxins concentration in the vehicle interior.

2. Photocatalytic reactor concept

Gas purifying can be obtained mainly in accordance with widely used physical and chemical processes. Well known is high-temperature oxidation (with or without catalyst) which occurs at temperatures of 200 to 1000°C – effective but not possible to apply in the vehicle ventilation system. The AVOCs can also be adsorbed on the active carbon, often used as a part of a cabin filter system. The capacity of the active carbon layer is relatively small. Most producers recommend to replace an active carbon filters every 6 months to keep its effectiveness. The alternative procedure for HC removal from inlet air is a photocatalytic oxidation (PCO) which is more effective, environmental friendly and free from economic issues [4]. The most common catalytic agent is titanium dioxide, TiO_2 , due to its capability to degrade a wide spectrum of chemical individuals when illuminated with UV light or when in the vicinity of ultraviolet [5–8]. Therefore, it appears that the extent of undesirable, toxic volatile organic compounds removability is dependent upon the UV light length. Furthermore, the reactions can course differently with respect to humidity pressure and temperature [9].

The concept of the reactor was created in Automotive Engineering Division on Mechanical Faculty of Wrocław University of Technology for Dr. Schneider Automotive Poland and is patented as the Vehicle Air Supply System (WO2015PL50072 20151221).

The system, comprising a light source and a photocatalyst between air intake and at least one supply port inside the system wherein the photocatalyst is located, is equipped with elements disturbing a flow of air stream, creating a network of ducts formed by a system of such that the time of exposure of the air stream to the photocatalyst is extended allowing for better mixing of the polluted air and time required for a photocatalytic process to reduce toxins [10].

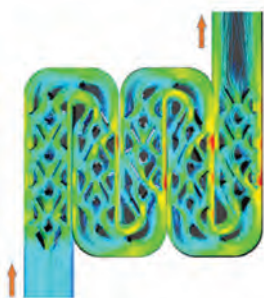


Fig. 1. Photocatalytic reactor concept [11]

The photocatalytic reactor tested here (shown in Fig. 2) was manufactured in additive technology (3D printing). Its constructional complicity was done for the purpose of polluted air disturbance and mixing. Moreover, detailed design and CFD computed channels were to elongate the time of exposure of the polluted air to catalytic agent, and prevent the negative influence of boundary layer occurrence.

The inserts are design in the way enable installation of a tape equipped with UV LEDs. In this case the length of UV light is 395 nm with power of 60 mW of each diode installed in the tape. The angle of illumination was 120 degrees.

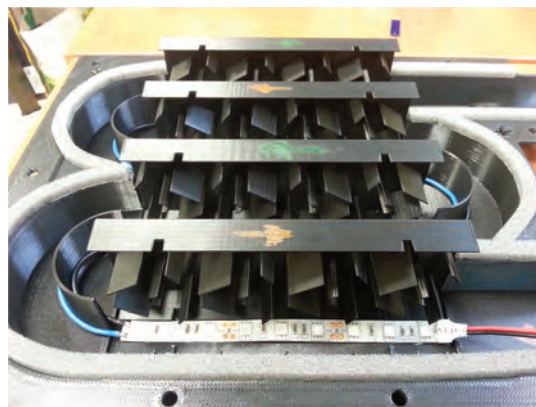


Fig. 2. Photocatalytic reactor used in the experiment

In this research as a photocatalytic agent titanium dioxide (TiO_2) was applied to the inserts of the reactor.

The applications procedures which were under consideration assumed varies techniques confronted with each other. The applications procedure described here results from long time investigation and were selected as the most assessable. For example thermal application of the catalytic agent was preliminary selected as one of the most effective procedures due to creation of constant and permanent layer of catalytic agent. However ultimately it was rejected due to significant difficulties and not efficient heating process.

Ultimately the mixture of a titanium dioxide with an adhesive substance was chosen for the investigation (see Fig. 3). The adhesive substance was mixed with a powder creating homogenous gel which was very easy distributed on the surface.

Mixing catalytic agent with the gel can cause immersion of the powder inside the gel which makes it not active. Only pollution contacting the active surface can be subjected to the photocatalysis. For this reason the experiment was repeated after fogging the surface, done in order to uncover layers of the catalytic agent. The results however showed that such action was not required as the not fogged surface was proven to be active



Fig. 3. Catalytic agent mixed with the adhesive substance

3. Research method

In order to simulate the effectiveness of the BTX reduction abilities of a tested reactor during real driving conditions, a test station was constructed. This test station comprises a test vehicle located in the climatic chamber for simulation of various weather conditions (e.g. temperature

and humidity). The photocatalytic reactor was assembled to the ventilation system of a vehicle in order to simulate its later implementation as a bypass to existing systems. During this studies it is assumed that the vehicle is ventilated in a closed loop – the atmosphere from the vehicle interior is aspirated by the system then warmed up or cooled and directed back again to the vehicle's cabin.

Before execution of the experiment, the cabin is set to be filled with pollutions (1:10 air diluted CI engine exhausts, emitted in repetitious conditions). The interior of the vehicle was deprived of most of the materials capable of accumulating the compounds related to exhaust gases directed to the cabin. During the experiment, the cabin of the vehicle is set to be inflated with pollutions provided form the power generator. After inflation, a photocatalytic reactor is engaged, and the BTX reduction process starts. The ventilation system is set to operate in a closed loop. Hence the crude air is directed through the reactor to the ventilation system and back again to the car cabin – as depicted in Fig. 4.

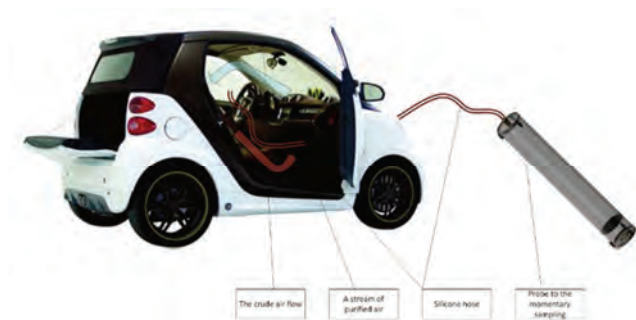


Fig. 4. Schematic representation of the experiment procedure [12]

Considering limited duration of the experiments along with limitation of aspirator flow rate it was necessary to adapt the test station for momentary sampling. The experiment assumed filling the cabin with the fumes and instant sampling in various times of the experiment. The chromatography tests requires specific amount of gases to be provided for analysis, otherwise the accuracy of the results is insufficient. This amount of polluted gas must be gathered instantly from the cabin. For this reason a testing vehicles was equipped in probe to the momentary sampling which aspirate specific amount of gases to the sampling bag. Limited time of the experiment exerted utilisation of at least two probes which gathers the polluted air one after the another allowing to draw characteristic of compound changing with respect to time. The probe is connected to the cabin via a silicon housing which aspirates the gas from the position of the drivers head

Air from the interior of the vehicle was momentary gathered by the probe to the bag for further chromatography tests. the probe is mounted outside the vehicle which requires to use a long silicon house. This in tur requires a large suction power for the bag to be instantly inflated with the polluted air. The exterior of testing vehicle is shown in Fig. 5.

The whole experiment is adjusted in order to consider full gas exchange during one cycle. At specific moments the atmosphere in the cabin was t sampled through the probe.

The test parameters like air flow, temperature and time sets was adjust to real drive conditions, in this research: temperature 290 K and 296 K, flow: 2 m/s, contact time 360 s (approximated time of one full circulation of the air through the system).



Fig. 5. Exterior of the tested vehicle with sampling probe

AVOCs Samples were collected by YEARS aspirator, model ASP-3 II, flow rate adjusted to 30 dm³/h, the amount of gas collected is 10 dm³. The gas was absorbed on active carbon Anasorb® SKC CSC. Activated carbon is poured into a glass tube of 5 cm³ and later it is emerged in 2 cm³ of carbon disulfide. The BTX in the samples was determined by gas chromatography according to the test procedure of the Emission Research Laboratory No. 1/2010 using the gas chromatograph Varian 450 – GC with flame ionization detector (FID), and a column Varian VF-WAXms 30 m × 0.25 mm ID DF: 0.25 μm. The work was performed at the set temperature of the column 373 K (110°C), the dispenser 523 K (250°C) and detectors 423 K (150°C). YEARS aspirator, model ASP-3 II, flow rate adjusted to 30 dm³/h, the amount of gas collected is 10 dm³, collected samples. The gas was absorbed on active carbon Anasorb® SKC CSC. Activated carbon is poured into a glass tube of 5 cm³ and later it is emerged in 2 cm³ of carbon disulfide. The glass tube is sealed with a stopper. Extraction takes place in a period of 20 min. After every few minutes the contents of the bottle was shaken in order to ensure adequate mixing of the material. Then 5 μl of the solution was gathered from above of the carbon layer. The gathered sample is injected into the chromatograph. Compounds designated as "residuals" have been converted to the concentration corresponding to n-pentan. In other words residuals are the compounds existing within the gas mixtures but was not identified by the chromatograph. The total relative error of the method was estimated at 20% (according to PN-EN ISO 16017-1: 2006).

4. Results

The results of the experiment are shown in Figs 3 and 4. Results of the experiment indicate an effective reduction of total volatile organic compounds (41% to 69%) in both temperatures. Despite the fact that there were visible differences between concentration reduction effectiveness in particular BTX compounds (especially xylenes), total BTX reduction is even slightly higher than total VOC (49% to 70%).

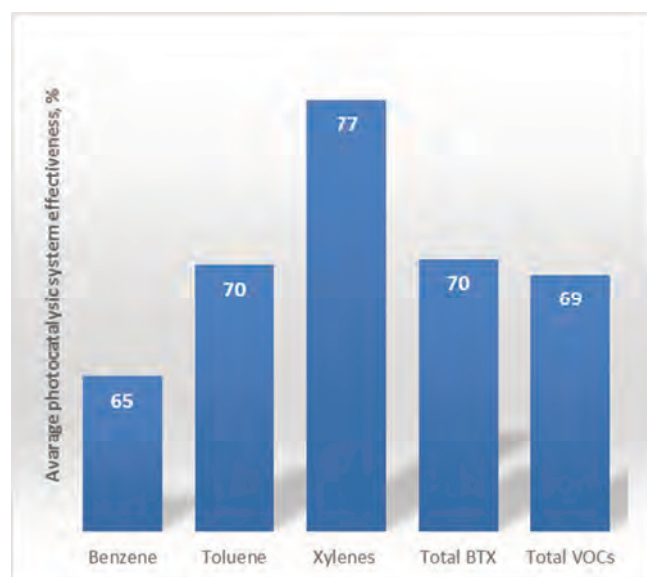


Fig. 3. Average photocatalytic system effectiveness, temperature 290 K

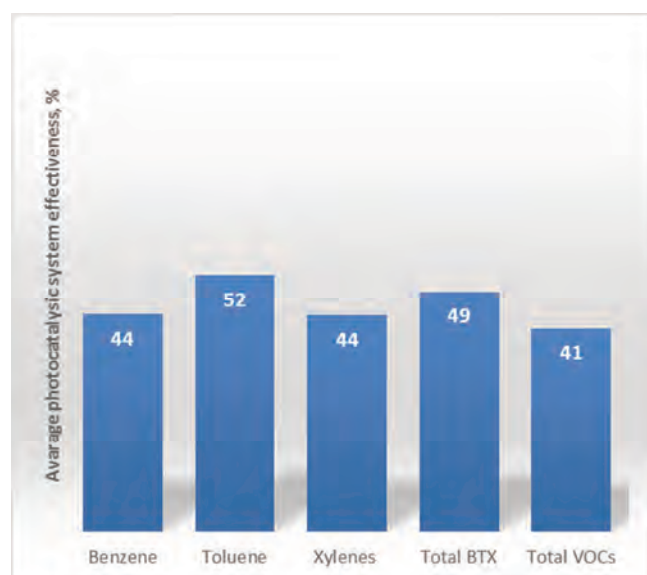


Fig. 4. Average photocatalytic system effectiveness, temperature 296 K

The experiment indicates that temperature can influence VOCs removal properties of the system, what also complies with other researches [11, 13]. The effect of the temperature increase on photocatalytic device effectiveness (Fig. 5) seems to be opposite to effectiveness of conventional catalytic reactors applied for IC engine exhausts cleaning (TWC, DOC), where the temperature increase significantly intensify HC conversion level (their effectiveness is minor in room temperature conditions).

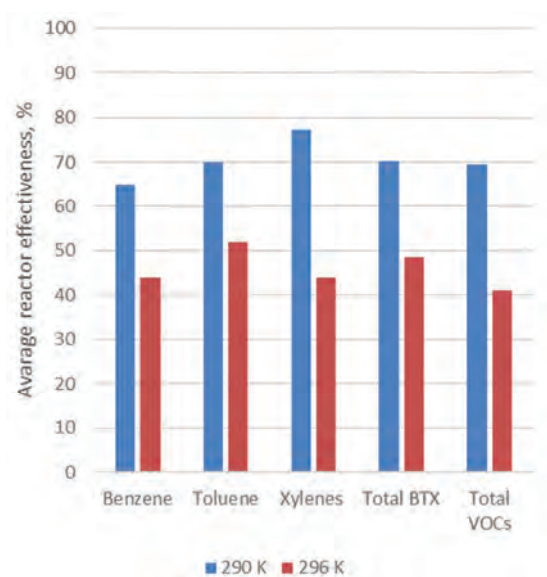


Fig. 5. Average photocatalytic system effectiveness, temperature comparison

5. Conclusions

1. IC Engine exhausts are a source of toxic aromatic volatile organic compounds. Especially benzene, toluene and xylene are substances responsible for many civilizational diseases (including cancer)
2. The BTX irritate nervous system which in consequence can be a direct influence on vehicle active safety (driver distraction, lowering driver cognitive abilities etc.).
3. Cooperation between Wroclaw University of Science and Technology and Dr. Schneider Automotive Poland effected of new product development: VOCs removal photocatalytic reactor for vehicle ventilation system.
4. The results of the experiment showed significant effectiveness of the device in BTX reduction (to 70%) in room temperatures.
5. The photocatalytic reactor seems to be prospective solution for air quality improvement in any vehicle cabin.
6. Taking the results of measurements into consideration, it appears that xylene in most cases is the most vulnerable to the catalytic agent purifying properties. Benzene showed opposite tendency. Its removability is in most cases the lowest amount the other compounds.
7. The BTX removal properties are highly dependent upon temperature, UV light length and the time of pollutant exposure to the catalytic agents. This justifies the attempt to improve the existing ventilation nozzles in terms of equipment related to BTX reduction.

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Nomenclature

IC internal combustion
VOCs volatile organic compounds
AVOCs aromatic volatile organic compounds
PCO photocatalytic oxidation
BTX benzene-toluene-xylenes

GC gas chromatography
FID flame ionization detector
TWC three way catalyst
DOC diesel oxidation catalyst

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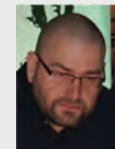
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The comparison of harmful substances emission from flex-fuel vehicle during NEDC and WLTC test cycles

The beginning of this article describes the NEDC and WLTC test cycles and basic differences between them. The following presents, the test stand and test objects, which were bioethanol fuels E10, E40 and E85. At the end were presented and discussed results of tests of these fuels in the flex-fuel vehicle in NEDC and WLTC test cycles.

Key words: flex-fuel, bioethanol fuels, NEDC, WLTC, WLTP

1. Introduction

Environmental protection is currently one of the main directions of technology development in the automotive industry. The aim is always to reduce emissions of harmful substances in vehicle exhaust gases. In the EU Commission Regulation No. 459/2012 of 29 May 2012, Euro emission standards specify the maximum emission of harmful components in exhaust gases, such as: NO_x, CO, HC, PM and PN in test cycles. In subsequent years, the maximum emission limits for these substances are constantly reduced. One of the ways to meet the rigorous standards is the use of alternative fuels for internal combustion engines. Bioethanol and its mixtures with motor gasoline are alternative fuels over which research is carried out on a wider scale [2, 6].

2. NEDC test cycle

NEDC (New European Driving Cycle), is a normalized test cycle, used to determine the content of harmful substances in the exhaust gases and the fuel consumption of combustion engines in passenger cars. This test cycle was supposed to imitate the conditions of driving a passenger car in a typical European city. Introduced in 1997, it was a mandatory test for car manufacturers to obtain approval of their vehicle in Europe. Complete NEDC cycle is a combination of four consecutive UDC (Urban Driving Cycle) cycles and one EUDC (Extra Urban Driving Cycle) cycle. For the first time, the UDC cycle was presented in 1970. It represents the behavior of a car moving in high-traffic. The characteristic features of the UDC cycle are: low engine load, relatively low exhaust gases temperature, driving speed not exceeding 50 km/h and frequent stops. The EUDC cycle was presented in 1990. It was supposed to represent a faster and more aggressive style of driving a passenger car, which is usually occurring in extra-urban conditions. The maximum vehicle speed in this cycle is 120 km/h.

Figure 1 shows the speed of the car during the entire NEDC test cycle divided into parts UDC and EUDC. Table 1 lists the basic parameters of the entire NEDC test cycle and its individual parts [1].

Table 1. Basic parameters of the NEDC test cycle and its parts UDC and EUDC test cycles [1]

	Unit	Test cycle		
		UDC	EUDC	NEDC
Distance	[km]	4052	6955	11007
Total time	[s]	780	400	1180
Average speed	[km/h]	18.7	62.6	34
Maximum speed	[km/h]	50	120	120

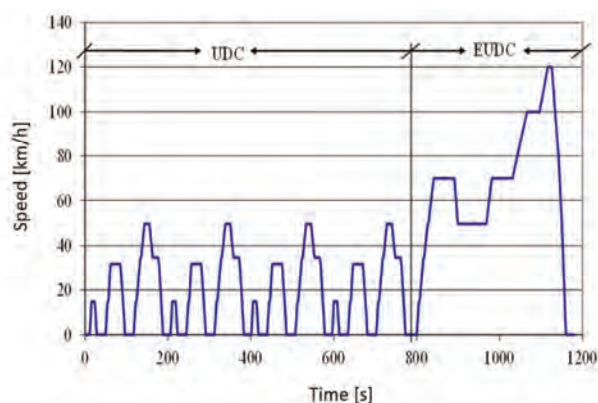


Fig. 1. The vehicle speed during NEDC test cycle [1]

3. WLTC test cycle

WLTC (Worldwide harmonized Light vehicles Test Cycles) is a test cycle designed to imitate real driving conditions for a car during a chassis dynamometer test. It is part of the procedure called WLTP (Worldwide Harmonized Light Vehicles Test Procedure) which is used to measure fuel consumption and emissions of harmful substances in the exhaust gases of passenger cars. The WLTP procedure was developed by the UN-ECE (United Nations Economic Commission for Europe) and published in 2014. Since September 2017, it replaces the current NEDC test cycle in the homologation process of new cars. There are several types of WLTC test cycles. The cycle type depends on the category to which the car is assigned on the basis of its PMR (Power to mass ratio) and the maximum vehicle speed specified in the manufacturer's specification. Table 2 shows the categories according to which the WLTP cycle type is determined along with the corresponding PMR coefficient values [1].

Table 2. WLTC test cycle categories with corresponding PMR coefficients and maximum speed values [1]

Category	PMR, W/kg	v_max, km/h
3b	PMR > 34	v_max ≥ 120
3a		v_max < 120
2	34 ≥ PMR > 22	–
1	PMR ≤ 22	–

The vast majority of cars in Europe qualify for category 3b. The test cycle of this category is divided into four phases. Each phase has different range of vehicle speed (small, medium, high, very high). Figure 1 shows the course of the car speed during the WLTC category 3b test cycle divided into individual phases.

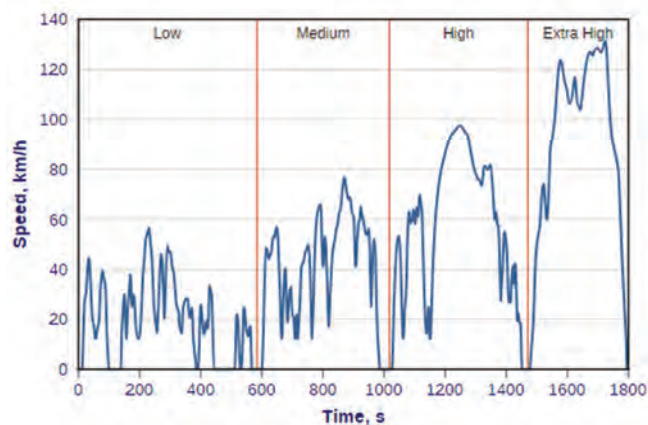


Fig. 2. The speed of the car during the WLTC category 3b cycle with the division into phases [1]

4. Test stand description

The tests were carried out on a chassis dynamometer at the Institute of Motor Transport in Warsaw. The research object was a Ford Focus passenger car. The vehicle has a spark-ignition engine with a displacement volume of 1798 cm³, factory equipped with a Flex Fuel system that allows it to operate on bioethanol fuels with a maximum bioethanol content of 85% (v/v). Before starting the tests, the vehicle's engine was warmed up to operating temperature.

The benchmark emission tests (Fig. 3) were carried out on a 2-roller chassis dynamometer 2PT220EX type, manufactured by Jaroš with two rollers with a diameter of 372 mm, with electric simulation of motion resistance and mechanical simulation of inertia placed in the low temperature chamber. This dynamometer allows the examination of vehicles (also at ambient temperature up to –14°C) with the following parameters:

- maximum net power on wheels up to 220 kW,
- maximum speed of 130/200 km/h,
- pressure on the drive axle up to 2400 kg,
- drive for one axle or more than one axis with the possibility of disconnecting the drive,
- maximum wheelbase of the drive axle: 2100 mm,
- maximum vehicle height 2900 mm, maximum distance of the rear drive axle from the front of the car 5000 mm.

The engine of the tested vehicle was powered by composed bioethanol fuels E10, E40 and E85. Table 3 presents the basic parameters of the mentioned fuels.



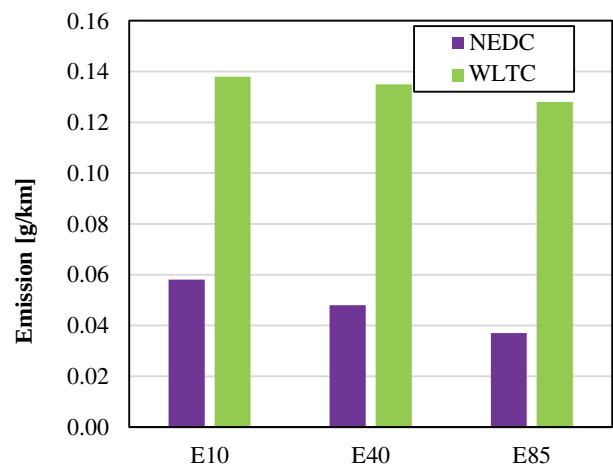
Fig. 3. The benchmark tests

Table 3. Basic parameters of E10, E40 and E85 fuels

	E10	E40	E85
Research octane number, RON	95	101	105
Heat value	42 MJ/dm ³	30.6 MJ/dm ³	22.7 MJ/dm ³
Density	730 kg/m ³	751 kg/m ³	782.2 kg/m ³
Bioethanol content (V/V)	10%	40%	85%

5. Test results

Figures 4 and 5 present the results of emission measurements of NO_x and CO respectively from the vehicle tested in NEDC and WLTC test cycles, the engine of which was powered sequentially by E10, E40 and E85. In the graphs, it can be observed that the higher content of bioethanol in the fuel makes that the tested vehicle emitted less NO_x and CO in the exhaust gases. In WLTC test cycle, reduction in emissions when using E85 fuel was about 7% for NO_x and about 13% for CO compared to E10 fuel. For both NO_x and CO, the emission value in the WLTC test cycle increases significantly compared to emissions in the NEDC cycle. The biggest change can be observed for NO_x emissions when fueled with E40, the value of this emission from the WLTC cycle increases by approx. 175% compared to the NEDC cycle.


Fig. 4. NO_x emission values from a vehicle with a SI engine powered by E10, E40 and E85 fuel in the NEDC and WLTC test cycle

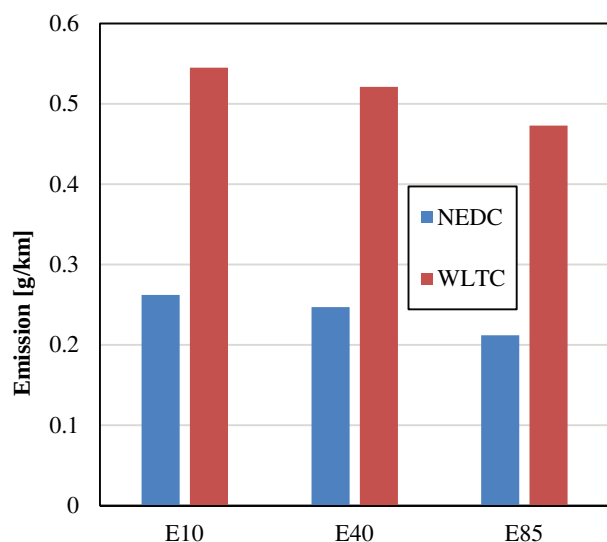


Fig. 5. CO emission values from a vehicle with a SI engine powered by E10, E40 and E85 fuel in the NEDC and WLTC test cycle

Figures 6 and 7 present the results of emission measurements of THC and CO₂ respectively from the vehicle tested in NEDC and WLTC test cycles, the engine of which was powered sequentially by E10, E40 and E85. Analyze of THC emission measurements has shown a reduction in THC emissions along with an increase in bioethanol content in the fuel. In the WLTC test cycle, with the E85 fuel, the THC emission was about 9% lower compared to the THC emission with the E10 fuel supply. The impact of bioethanol content in fuel on the value of CO₂ emissions was lower than in the emissions of previously discussed substances. In WLTC test cycle, with E85 fuel supply, the CO₂ emission value was about 5% lower compared to CO₂ emission values when fueled with E10 fuel. For each of the tested fuels in the WLTC test cycle, the THC emission value was significantly higher compared to the values of these emissions in the NEDC test cycle. For E85 fuel, THC

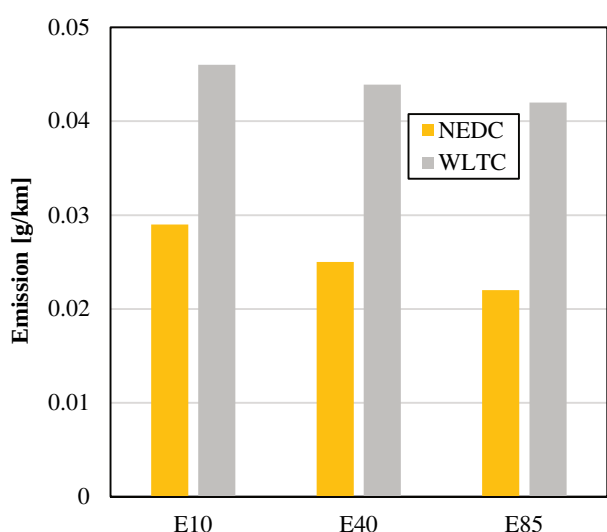


Fig. 6. THC emission values from a vehicle with a SI engine powered by E10, E40 and E85 fuel in the NEDC and WLTC test cycle

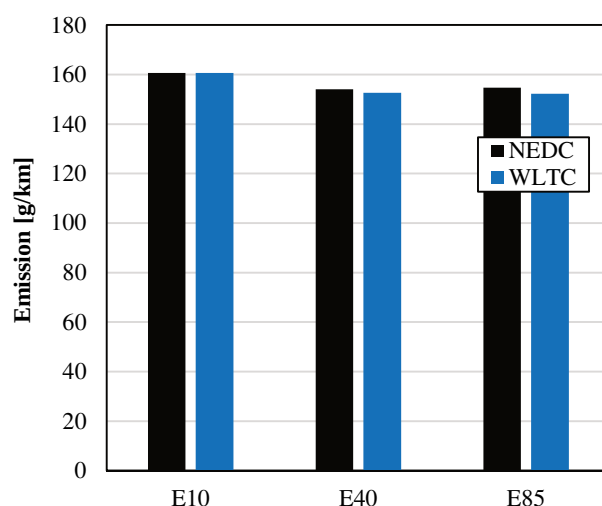


Fig. 7. CO₂ emission values from a vehicle with a SI engine powered by E10, E40 and E85 fuel in the NEDC and WLTC test cycle

emission in the WLTC test cycle was about 91% higher than the THC emissions in the NEDC test cycle. In the case of CO₂ emissions, the type of test cycle did not have a significant impact on their values.

Figure 8 shows the consumption values of E10, E40 and E85 fuels by the vehicle engine tested in NEDC and WLTC cycles. The higher content of bioethanol in the fuel caused its higher consumption. Such regularity occurred both in the NEDC and WLTC test cycles. The difference between consumption of E10 fuel and consumption of E85 fuel in the WLTC test cycle was about 37%. The results of fuel consumption measurements in the WLTC test cycle have higher values than in the NEDC test cycle for the corresponding fuels.

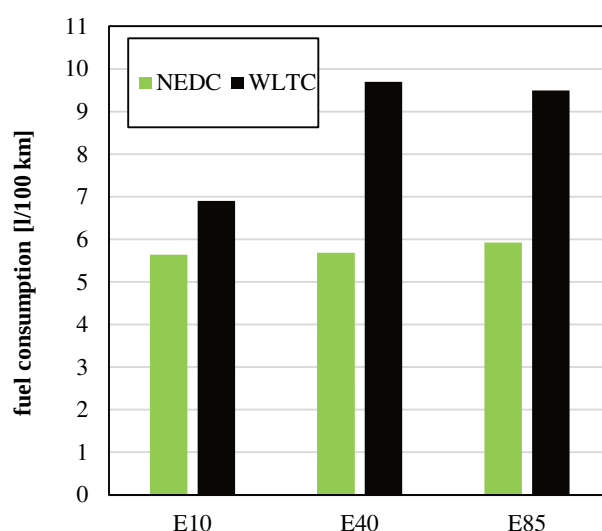


Fig. 8. Fuel consumption of a vehicle with a SI engine powered by E10, E40 and E85 fuel in the NEDC and WLTC test cycle

6. Summary

On the basis of the conducted research, it is possible to notice a significant (even over 100%) increase in NO_x, CO and THC emissions in the WLTC test cycle in comparison

to the emission values of these substances in the NEDC test cycle. This is due to the fact that the tested vehicle in the WLTC test cycle reaches higher maximum speeds and the speed change is more dynamic. Instead, CO₂ emission values were compared in both of these test cycles.

Fuel consumption measurements of the vehicle's engine in the WLTC test cycle showed higher values compared to the fuel consumption in the NEDC test cycle. The largest difference occurred for E40 and E85 fuels, where the increase in their consumption in the WLTC cycle amounted

to approx. 70% and approx. 61% respectively in comparison to consumption in the NEDC cycle.

The impact of bioethanol in fuel on measured emissions of exhaust gases was similar in both the NEDC and WLTC test cycles. With increase of bioethanol content in fuel, a distinct reduction in NO_x, CO and THC emissions was observed. The value of CO₂ emission in exhaust gases decreased minimally for engine powered by E40 and E85 fuels.

Nomenclature

EUDC	Extra Urban Driving Cycle
NEDC	New European Driving Cycle
PMR	Power to mass ratio
UDC	Urban Driving Cycle
UN-ECE	United Nations Economic Commission for Europe

WLTC	Worldwide harmonized Light vehicles Test Cycles
WLTP	Worldwide Harmonized Light Vehicles Test Procedure

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Cold start emissions of passenger cars with gasoline and diesel engines in Real Driving Emissions tests

Modernization of passenger cars and constant development of existing legislation lead to a reduction of exhaust emissions from these vehicles. In accordance with package 3 of the RDE test procedure, the European Commission has extended testing methods by including exhaust emissions during a cold start. The article compares the research results on the impact of ambient temperature during the cold start of spark-ignition and compression-ignition engines in road emission tests. The tests were carried out in line with the requirements of the RDE test procedure for passenger cars meeting the Euro 6d-Temp emissions standard. The obtained results were analyzed, i.e. there were compared the engine and vehicle operation parameters and the values of road exhaust emissions, during the cold start of gasoline and diesel engines at the ambient temperature of approximately 25°C. The summary presents the share of cold start phase of a passenger car (at the ambient temperature of around 25°C) for each exhaust emission compound in the urban part of the test, and in the entire RDE test, depending on the engine type used.

Key words: exhaust emission, passenger cars, Real Driving Emissions

1. Introduction

Despite constant changes in legislation, road transport is still considered the most important source of hydrocarbons, carbon monoxide, carbon dioxide, nitrogen oxides, and particulate matter emissions. These compounds cause not only the deterioration of air quality but also the human health condition. This effect is particularly visible in many urban areas. It has led legislators to adopt more stringent pollutant emission standards for the automotive sector worldwide [17]. For example, by 2021 the limits on carbon dioxide emissions will have been reduced to 95 g/km. Unfortunately, the growing number of passenger cars moving within the European Union is becoming a real issue. That is the reason to develop better key emission control strategies.

The analysis of the cold start emissions level is particularly important because it is perceptible mainly in urban areas where it is the immediate vicinity of people. In recent years, many publications about the impact of ambient temperature on exhaust emissions during a cold start have been published. The impact of low ambient temperatures on pollutant emissions in lab tests has been described in detail. Already in 2011, the authors of the article [1] noticed the problem of cold start at low ambient temperatures. The objects of their research were both gasoline and diesel engines meeting the Euro 4 and Euro 5 emission standards in the NEDC test (at ambient temperatures of 24°C and -7°C). They pointed out that if the oil and coolant temperatures are equal or close to ambient temperature, starting up an engine, with no apparent increase in pollutant emissions and increased fuel consumption, may be more difficult to obtain (especially at low ambient temperatures). The authors of the article [16] came to the same conclusions but for the next generation of emission standards, i.e. Euro 6. The authors of article [14] determined the impact of various ambient temperatures (-7°C and 23°C) on the emission results of vehicles equipped with compression and spark ignition engines. The aim of the test was to verify the results according to Euro 6 standards and check if the exhaust emis-

sions in WLTC test increase during the cold season in comparison to higher ambient temperatures. The authors focused on determining the effect of an ambient temperature on exhaust emissions in the entire WLTC test, and not only during the cold start period. The research results indicated that the tested vehicles (at the ambient temperature of 23°C) were in line with type approval of Euro 6 vehicles (still using NEDC), with the exception of NO_x emissions from diesel vehicles that were 2.3–6 times higher than Euro 6 standards. Lowering the ambient temperature to -7°C caused a significant increase in exhaust emissions from the tested vehicles. The largest differences in the CO and THC emission values were observed for the spark-ignition engine at two different ambient temperatures. The authors of this article noticed the need to develop a new and independent procedure allowing verification of the emissions level at low ambient temperatures.

It is worth noticing that approval tests carried out in laboratory conditions (on a chassis dynamometer) are not sufficient. They have been introduced as a complementary exhaust emissions testing in real traffic conditions [10, 13]. The research on exhaust emissions from transport vehicles in conditions of real operation reflect the actual ecological performance of vehicles [7–9]. Their results showed an increased emission of some exhaust components and particulates in comparison to tests carried out in laboratory conditions [12, 15]. The authors of the publication [11] determined the impact of two different ambient temperatures (8°C and 25°C) during the cold start of a spark-ignition engine in emission road tests. They defined the share of the cold start phases for different ambient temperatures in the entire RDE test, but also in the urban part of the RDE test. The share of the cold start in the urban part of RDE test is the highest at the temperature of 8°C. The values are as follows: CO₂ (11.6%) and 3.4% (the entire RDE test), CO (26%), NO_x (10.6%), PM (22.7%). The share of the cold start at the temperature of 25°C in the urban part of

the RDE test is as follows: CO₂ (9.8%), CO (10.6%), NO_x (12.3%), PM (18.4%).

This article is an extension of the publication [11]. The aim of this article is to compare the share of a cold start phase of a passenger car (at one ambient temperature – around 25°C) for each exhaust emission compound in the urban part of the test, and in the entire RDE test, depending on the engine type used (gasoline or diesel).

2. Requirements of RDE tests

The RDE test needs to meet a lot of requirements which have listed in Fig. 1. The test is divided into three sections: urban, rural and motorway conditions. The test drive is carried out continuously. The rural drive section can be interrupted by short periods of urban driving if urban areas are on the route. Driving on the motorway can interrupt short driving periods in urban or rural areas. The duration of the test is within the acceptable range, i.e. from 90 minutes to 120 minutes.

REQUIREMENTS	URBAN	RURAL	MOTORWAY
VEHICLE SPEED	0–60 KM/H	60–90 KM/H	90–145 KM/H
DISTANCE SHARE	29–44%	23–43%	23–43%
MIN DISTANCE	16 KM	16 KM	16 KM
AVERAGE SPEED	15–40 KM/H	–	–
TOTAL STOP TIME	6–30%	–	–
INDIVIDUAL STOP TIME	≤ 300 S	–	–
V > 100 KM/H	–	–	≥ 5 MIN
V > 145 KM/H	–	–	< 3% M _{TIME} *
TOTAL TEST DURATION: 90–120 MIN			
START/END TEST ELEVATION DIFFERENCE:			
*M _{TIME} – MOTORWAY TIME			

Fig. 1. The RDE test requirements [2–5]

Package 3 of the RDE test procedure have introduced among others the inclusion of cold start period emissions in the whole RDE test. Cold start is the period from the first start of combustion engine until the point when the combustion engine has run cumulative for 5 min. The cold start period ends once the coolant temperature has reached 70°C for the first time but no later than after 5 min from initial engine start [6]. The requirements for cold start are listed in Fig. 2.

REQUIREMENTS FOR COLD START
MAX VEHICLE SPEED 120 KM/H
AVERAGE VEHICLE SPEED (INCLUDING STOPS) 15–40 KM/H
TOTAL STOP TIME < 90 S
IDLING AFTER IGNITION 15 S
IDLING AFTER VEHICLE CONDITIONING FOR COLD START TESTING DRIVEN FOR AT LEAST 30 MIN FOLLOWED BY SOAK DURATION IN THE RANGE OF 6 TO 56 HOURS

Fig. 2. Cold start requirements [6]

3. Research methodology

The study consisted of the pollutant emissions comparison and describing the relations between test results at the ambient temperature of 25°C. Before the road test, the

vehicles were conditioned at ambient temperature for 24 hours. Accordingly, the engine oil and engine coolant temperatures corresponded to ambient temperature, so the requirement of a cold engine start (taking into account the appropriate ambient temperature) was met.

Research objects

During the tests, there were used passenger cars of the same brand with the curb weight of approximately 1450 kg and Euro 6d-Temp emissions standard (vehicles approved in accordance with the WLTP procedure). The first one was equipped with a gasoline engine and the second one with a diesel engine. Both engines were turbocharged. The engine displacement was 1.6 dm³, the maximum power was 130 kW and 125 kW respectively, and the maximum torque was 300 Nm and 350 Nm respectively. In the case of the gasoline engine, the exhaust aftertreatment system was a three-way catalytic converter with a particulate filter, and in the case of the diesel engine, there were an oxidation reactor, a selective catalytic reduction system and a particulate filter. The choice of such research objects was dictated by the fulfillment of requirements relating to RDE tests.

Research apparatus

Emission measurements were carried out in real driving conditions; this approach requires the installation of the gas sampling apparatus on the vehicle in such a way that ensures its normal operation. Therefore, a gas sampling system was prepared, which together with the system measuring the flow rate of the exhaust gas also performed partial sampling of the flue gas for the analyzers to enable making the measurement (Figure 3 shows the wiring schematic of the measuring equipment).

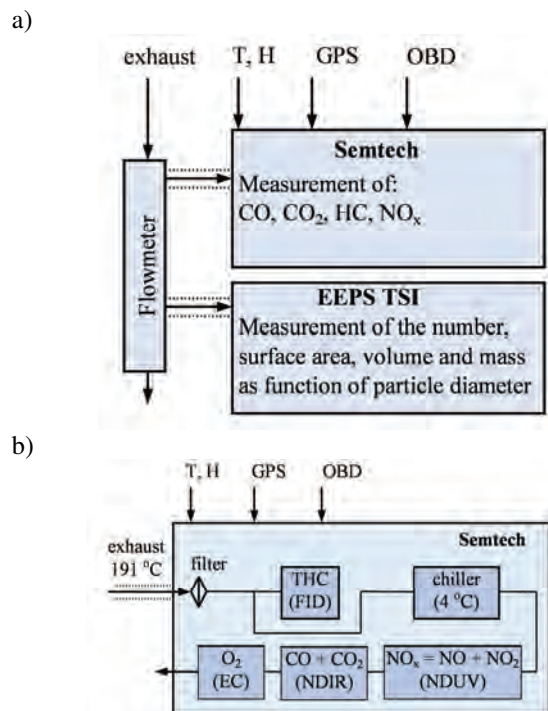


Fig. 3. Diagram of the measurement system used for testing (a) and mobile exhaust gas analyzer with marked flue gas flow (b) for the measurement of gaseous compounds and particulates; T – ambient temperature, H – air humidity

Research route

The research route was designated in accordance with the RDE requirements and divided into 3 parts: urban, rural and motorway (Fig. 4). The lengths and shares of individual parts were in line with the requirements specified in EU standards EU 2016/427 [3] and EU 2016/646 [4]. The test distance was about 90 km, and the average velocity during the test was around 45 km/h. The test was repeated twice for two vehicles with different type of engine (gasoline and diesel).

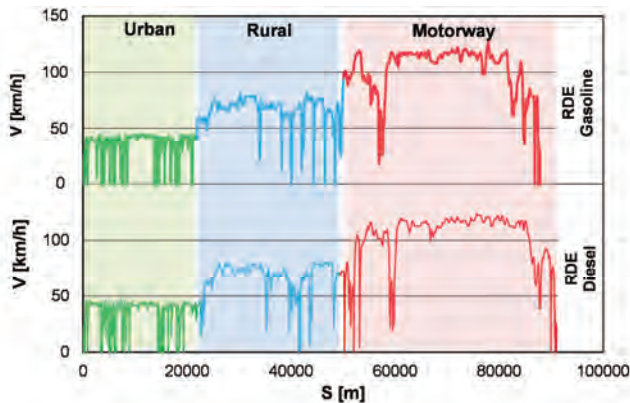


Fig. 4. Velocity of individual test drives

4. The analysis of recorded parameters

Operating parameters of an engine (a vehicle)

The first stage of the study was to check the validity of performing road tests, which are specified in the Regulations [3, 4, 6], mainly concerning the length of individual road test phases and shares in the entire test, the test duration, the share of stops in the urban part, and driving dynamics in particular parts of the RDE test. All parameters were verified by appropriate procedures and no deviations from the required values were found (Tab. 1). For a reliable

comparison, there was also carried out an analysis regarding the similarity of test drives.

Therefore, the characteristics of the individual driving stages duration in the velocity–acceleration co-ordinates for vehicles equipped with a spark-ignition engine and a compression-ignition engine were made. This comparison shows (Fig. 5) that there is a large similarity of test drives.

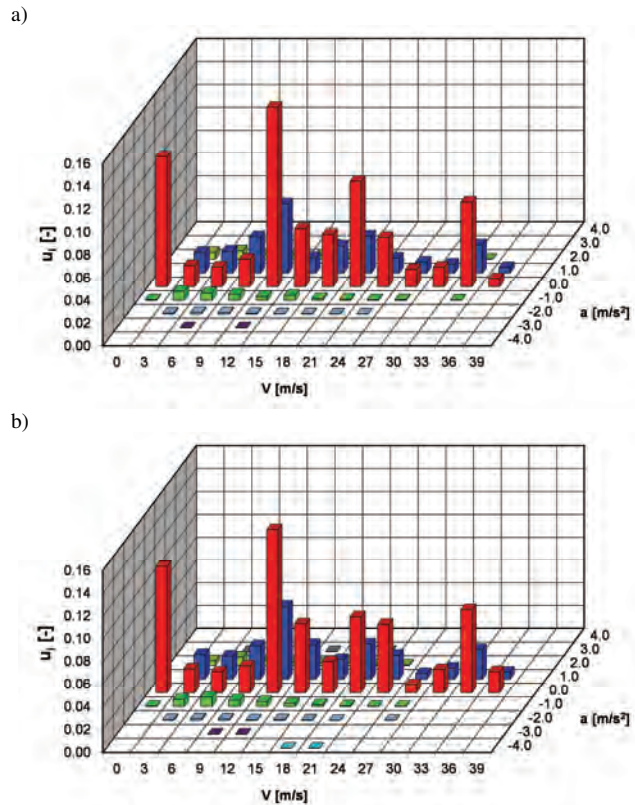


Fig. 5. The characteristics of vehicle operating time share in the velocity–acceleration co-ordinates for gasoline engine (a) and diesel engine (b)

Table 1. Validation of trip characteristics in accordance with EU RDE requirements

Trip characteristics	Gasoline	Diesel	Requirement	Valid?
Urban distance [km]	29.9	30.5	> 16	OK
Rural distance [km]	28.1	27.9	> 16	OK
Motorway distance [km]	29.9	32.6	> 16	OK
Total trip distance [km]	87.9	91.0	> 48	OK
Urban distance share [%]	34.1	33.5	29–44	OK
Rural distance share [%]	31.9	30.7	33 ± 10	OK
Motorway distance share [%]	34.0	35.8	33 ± 10	OK
Urban average speed [km/h]	27.6	27.6	15–40	OK
Urban stop time [%]	19.8	19.1	6–30	OK
Motorway speed above 100 km/h [min]	13.4	16.0	> 5	OK
Motorway speed max [km/h]	127.2	123.8	< 160	OK
Motorway speed above 145 km/h [%]	0.0	0.0	< 3	OK
Total trip duration [min]	104.7	106.6	90–120	OK
Cold start requirements				
Cooling temperature [°C]	67.0	59.0	< 70	OK
Max vehicle speed [km/h]	41.9	46.7	< 60	OK
Stop time [s]	36	37	< 90	OK
Idling after ignition [s]	8	6	< 15	OK
Overall trip dynamics				
Urban 95th percentile $V \cdot a_+$ [m^2/s^3]	12.9	12.3	< 18.198 / < 18.228	OK
Rural 95th percentile $V \cdot a_+$ [m^2/s^3]	13.8	17.2	< 24.166 / < 24.361	OK
Motorway 95th percentile $V \cdot a_+$ [m^2/s^3]	15.4	14.7	< 27.194 / < 27.304	OK
Urban RPA [m/s^2]	0.14	0.15	> 0.131 / > 0.131	OK
Rural RPA [m/s^2]	0.070	0.073	> 0.061 / > 0.059	OK
Motorway RPA [m/s^2]	0.070	0.048	> 0.025 / > 0.025	OK

The possibility of comparing the results of exhaust emissions in road tests is valid only in the case of test drives similarity. Road tests are characterized by high uniqueness of road conditions, despite meeting the formal requirements presented above. The definition of test performance correctness, described by relative positive acceleration or the product of vehicle velocity and positive acceleration, may not be sufficient to confirm the comparability of results obtained in road tests.

Comparison of the vehicle operating conditions can be made on the basis of two-dimensional characteristics of vehicle operation. For this purpose, the relevant shares of vehicle operating time are compared on the two-dimensional characteristics. In this way, two data columns are obtained that will be used to determine the regression equation ($y = ax + b$). The determination coefficient (R^2) is a measure on the basis of which the test drives can be compared. The compared data is similar if the straight-line coefficient is close to 1 (assuming that the value of constant term is equal to 0). Such comparison is presented in Fig. 6, where the entire RDE test, urban, rural and motorway sections of RDE test are collated (for the vehicle with spark-ignition engine and the compression-ignition engine). The obtained values of the determination coefficient are close to the value of 1, which means high compatibility between the obtained operating conditions of both vehicles in RDE tests.

However, the most important was to check the correctness of the cold start parameters and initial conditions of the road test start (these parameters are discussed in the introduction). For this reason, the vehicle velocity analysis during the cold start period (Fig. 7) has been made. It is required to start driving after a maximum of 15 seconds from starting the engine. This condition was met in both cases and at the same time a similar character of the vehicle velocity changes is observed in the first 300 s of the RDE (slightly higher values of the average vehicle velocity were recorded for the vehicle equipped with a diesel engine).

The comparison of the coolant temperature in both cases also shows that the requirements of the RDE test procedure have been met, as it did not exceed 70°C. During the gasoline engine tests, 300 s from the engine start, the coolant temperature was 67°C, and for the vehicle fitted with the diesel engine the temperature was only 62°C. The initial temperature difference between the engine coolant in both type of engines was at the value of 1°C, and after 300 s (since the engine was started) it increased to 5°C (Fig. 8).

A similar character of changes was also observed during the analysis of exhaust temperature, measured at the end of the vehicle exhaust system (Fig. 9). The difference occurring for the measurement time ($t = 0$) of around 3°C is compensated already after 20 s, where the exhaust temperature is about 20°C. The further course of exhaust temperature changes (up to $t = 200$ s) is very similar to each other (exhaust temperature around 40°C) and the occurring differences in this period are very small ($\pm 5^\circ\text{C}$). However, the last 100 s of the period are different – for the gasoline engine the exhaust temperature rises to 47°C, and for the diesel engine it decreases to 36°C.

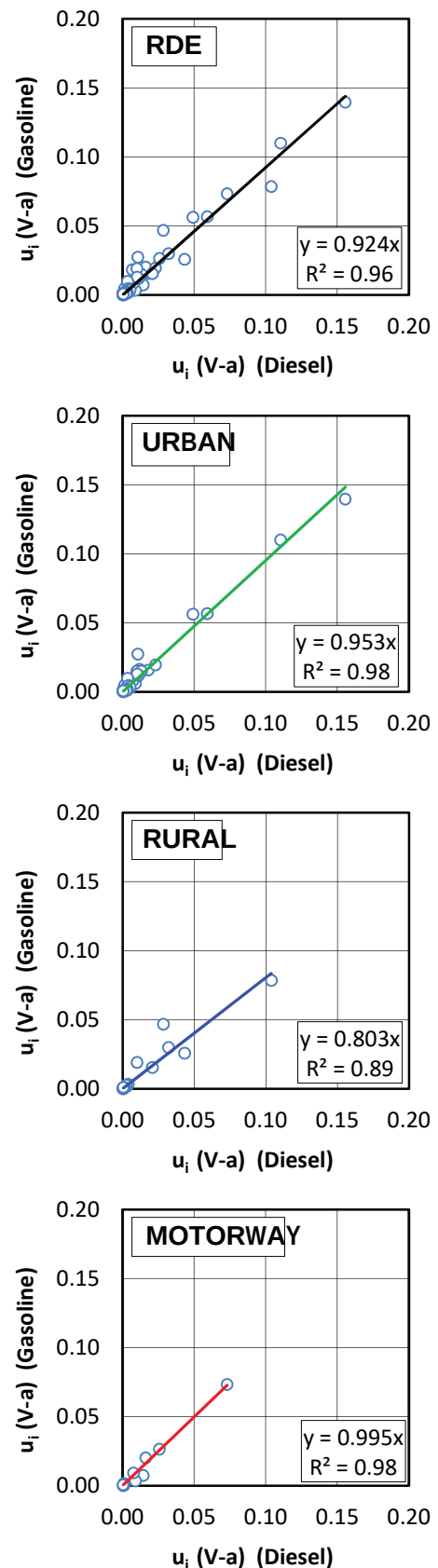


Fig. 6. Comparison of the test drives compliance (vehicles with gasoline engine and diesel engine) with the use of linear correlation of vehicle operating time share in each velocity and acceleration compartment

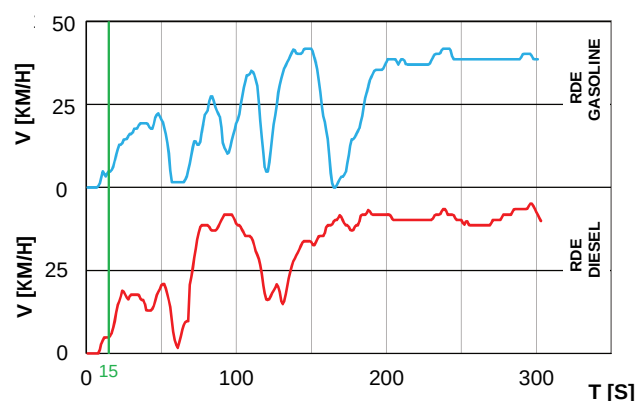


Fig. 7. The vehicles velocity during the phase of the engine cold start

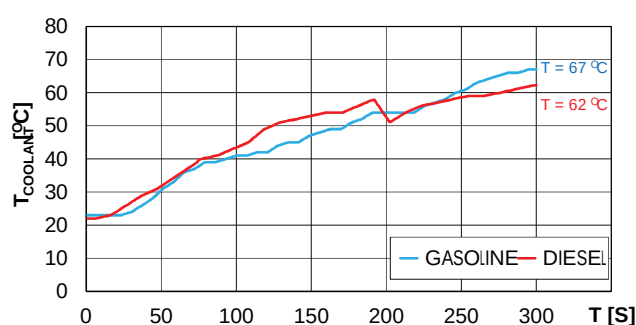


Fig. 8. The coolant temperature during the cold start phase of the gasoline and diesel engines

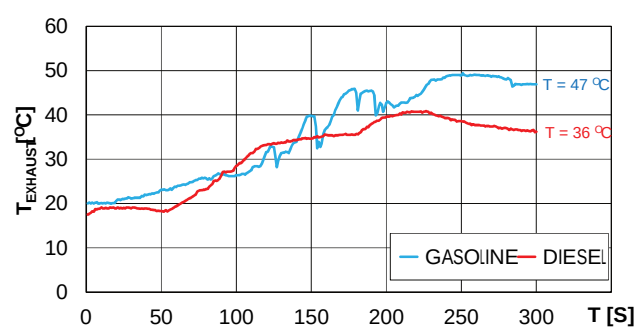


Fig. 9. The exhaust temperature during the cold start phase of the gasoline and diesel engines

The presented analysis of selected engine operating parameters allowed to find the similarity of engine operation conditions (and vehicles), and the element differentiating both measurements is the type of engine used in tests. On this basis, it was reasonable to take further actions that will answer the question about differences in concentration and emission of exhaust compounds in the initial period of vehicle operation (equipped with a different type of combustion engine).

Ecological parameters

The recorded course of exhaust concentration changes in the first period ($t = 0\text{--}300$ s) of the cold start phase for gasoline and diesel engines are presented below.

The analysis of carbon dioxide concentration (Fig. 10) shows that during the cold start and warming the engine,

the average concentration of this compound was about twice smaller for the compression-ignition engine (7.6%) compared to the spark-ignition engine (14.3%). Slight fluctuations from the value set for the spark-ignition engine were the result of a change in the vehicle velocity – changes in the dynamic motion conditions. However, the average value of this parameter is not affected by the ambient temperature at which the cold engine started. For the compression-ignition engine, the course of changes in the carbon dioxide concentration was in the range of 4% to 12%, and resulted mainly from the change of the instantaneous engine load.

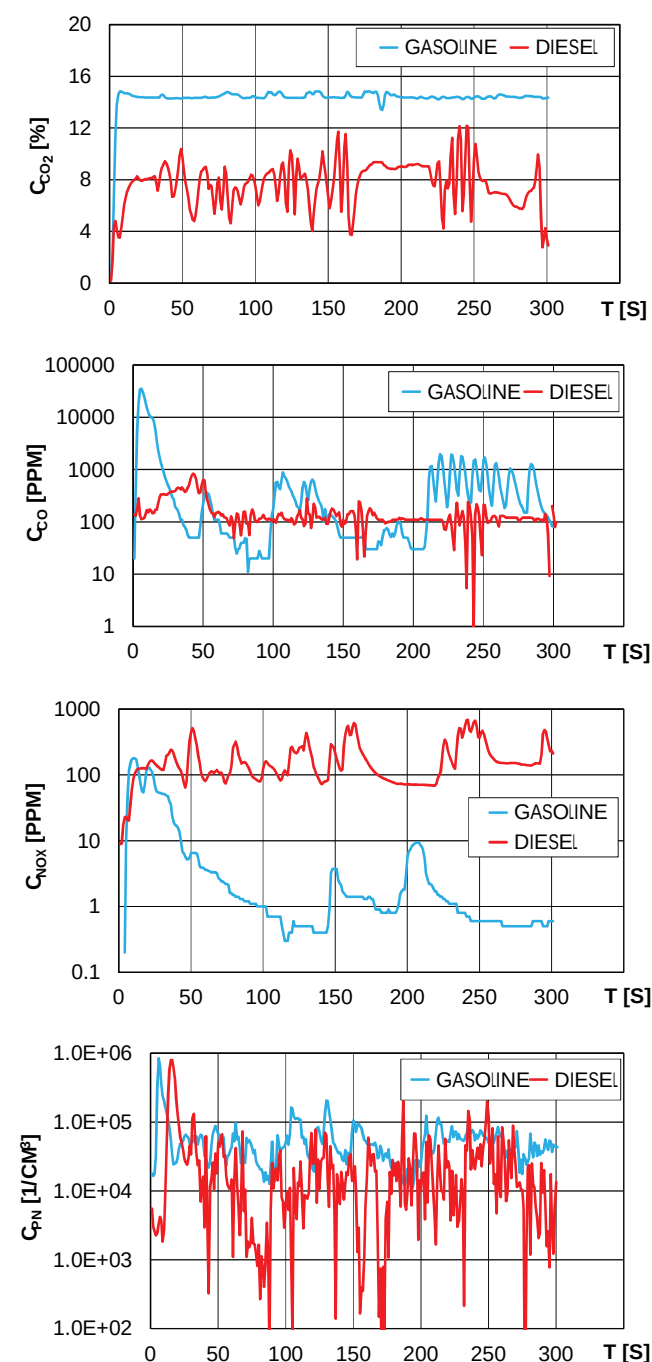


Fig. 10. The change of exhaust gases concentration during 300 s from the cold start for gasoline and diesel engines: carbon dioxide, carbon monoxide, nitrogen oxides, and particles number

The course of changes in the carbon monoxide concentration is different in nature. The characteristic instantaneous increase in the concentration during the cold start phase of gasoline engine reaches the value of about 30,000 ppm, however it drops very quickly to 50 ppm. This reduction occurs in the value of about 50 s. Then there is a change in concentration in the range of 100–1000 ppm caused mainly by the change of the engine load. In the case of diesel engine cold start, the values of carbon monoxide concentration are much lower. The first increase of this compound after engine start is around the value of 1000 ppm, and then the concentration oscillates about the value of 100 ppm. Therefore, the carbon monoxide emission intensity should increase significantly in the first period of the gasoline engine cold start phase in comparison to the diesel engine.

The concentration of nitrogen oxides is different from the analysis of the previous compound. For the gasoline engine cold start phase, an increased concentration of nitrogen oxides (up to 100–150 ppm) is observed only in the initial phase (within the first 50 s after starting the engine). In the 50th second of the RDE test, the value of nitrogen oxides concentration for the gasoline engine is 10 times lower than for diesel engine. After time of 100 s, the concentration of nitrogen oxides is 100-fold lower for gasoline engine in comparison to diesel engine. For the diesel engine, the concentration of NO_x was remained at the level of 100–1000 ppm throughout the whole period considered, while for the gasoline engine it did not exceed 10 ppm.

Regarding the particles number, there were observed higher values for the gasoline cold start ($1.0 \cdot 10^4$ $1/\text{cm}^3$ – $1.0 \cdot 10^5$ $1/\text{cm}^3$) in comparison to the diesel engine ($1.0 \cdot 10^2$ $1/\text{cm}^3$ – $1.0 \cdot 10^5$ $1/\text{cm}^3$). The average value of the particulate matter concentration for the gasoline engine in the considered period of 300 s was $5.9 \cdot 10^5$ $1/\text{cm}^3$, and for the diesel engine $3.4 \cdot 10^4$ $1/\text{cm}^3$. After about 230 s from the cold start, the number concentration of particles in both cases stabilized at $1 \cdot 10^4$ $1/\text{cm}^3$.

Analysis of test results regarding the engine cold start phase

Taking into account the exhaust mass flow rate, a quantitative evaluation of the exhaust emissions was made in the period of 300 s from the starting of the gasoline and diesel engines (Fig. 11). In the case of carbon dioxide mass, it is higher by 2 times (after 300 s from engine start) for the gasoline engine than for the diesel engine. When the measurement was related to the gasoline engine, the carbon dioxide mass was 677 g, for the diesel engine the mass was equal to 312 g.

The carbon monoxide mass during the 300 s period of the gasoline engine cold start was 3.45 g, and for the diesel engine it was 23 times lower and amounted to 0.14 g. The nitrogen oxides mass was more than 16 times higher when the diesel engine started (1.103 g) than when started the gasoline engine (0.034 g). The smallest difference in the measurement of exhaust compounds for two different engines occurred when comparing the number of particles. In this case, it was found that after the engine operating time from the cold start phase of 300 s, the larger number of particles was emitted by a vehicle equipped with a diesel

engine ($2.2 \cdot 10^8$), for the gasoline engine the value of particles number was $1.4 \cdot 10^8$.

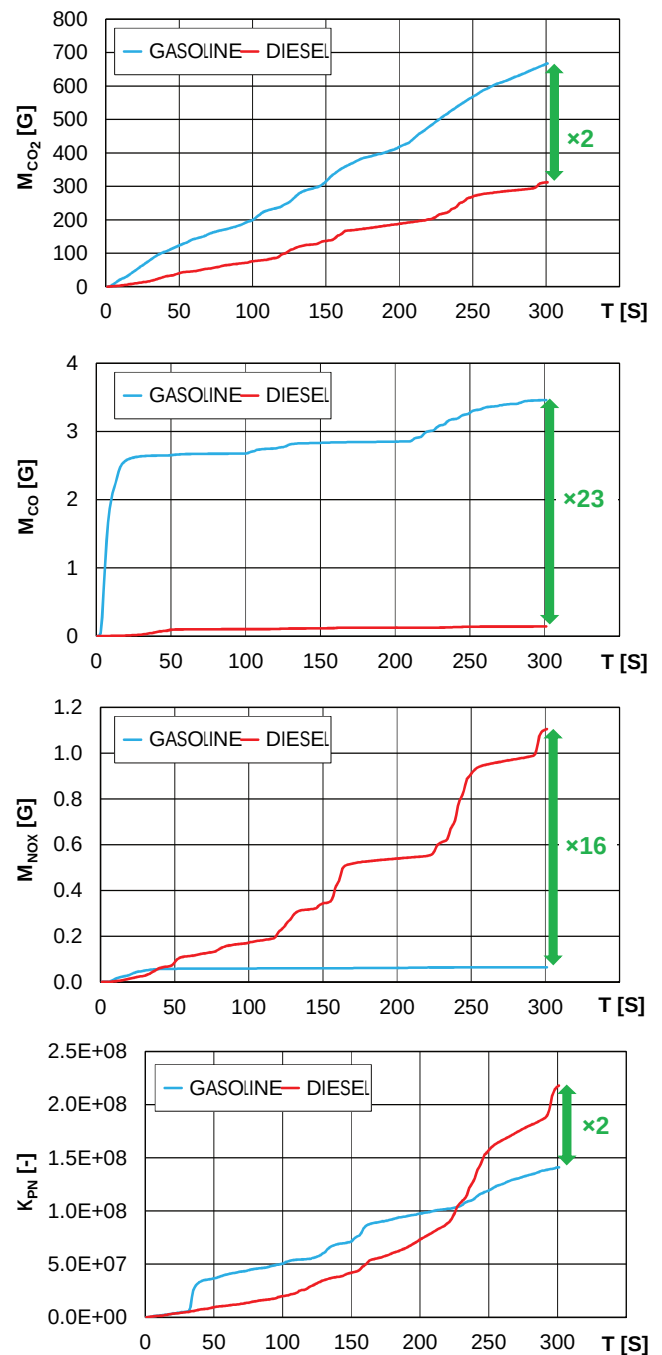


Fig. 11. The mass of carbon dioxide, carbon monoxide, nitrogen oxides and the number of particles in the cold start period of 300 s for gasoline and diesel engines

The presented data were used to determine the road emissions (converting the mass per unit of the road) during the cold start phase (lasting 300 s) and the entire urban phase.

When comparing the results of the exhaust emissions to a vehicle equipped with a gasoline engine, it follows that (Fig. 12):

- carbon dioxide road emissions in the cold start phase (304 g/km) is higher by 51% than in the entire urban phase of the RDE test (202 g/km),
- carbon monoxide road emissions in the cold start phase (155 mg/km) is higher by 64% than in the entire urban phase of the RDE test (95 mg/km),
- nitrogen oxides road emission in the cold start phase (20.5 mg/km) is higher by 89% than in the entire urban phase of the RDE test (10.9 mg/km),
- the number of particles in the cold start phase ($1.5 \cdot 10^{10}$ 1/km) is higher by 182% than in the entire urban phase of the RDE test ($5.2 \cdot 10^9$ 1/km).

The changes in road emissions of individual exhaust compounds during the cold start phase of the diesel engine in relation to the whole part of the urban phase of RDE test are as follows (Fig. 13):

- carbon dioxide road emissions is higher by 70% (341 g/km and 200 g/km respectively),
- carbon monoxide road emissions is lower by 12% (119 mg/km and 135 mg/km respectively),
- nitrogen oxides road emission is higher by 7% (42.3 mg/km and 39.5 mg/km respectively),
- number of particles is higher by 82% ($5.1 \cdot 10^{11}$ 1/km and $2.8 \cdot 10^{11}$ 1/km respectively).

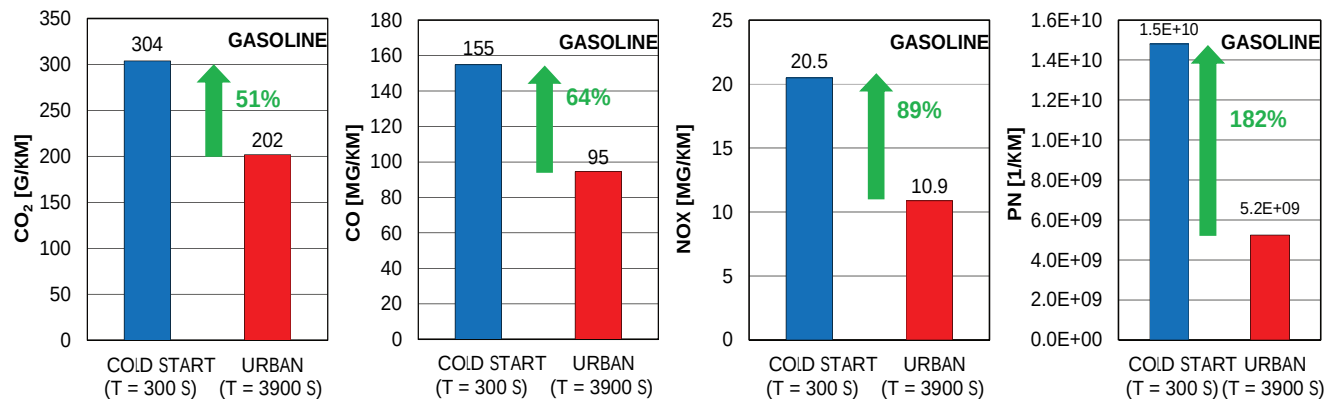


Fig. 12. Road emissions of carbon dioxide, carbon monoxide, nitrogen oxides and particles number during cold start of gasoline engine

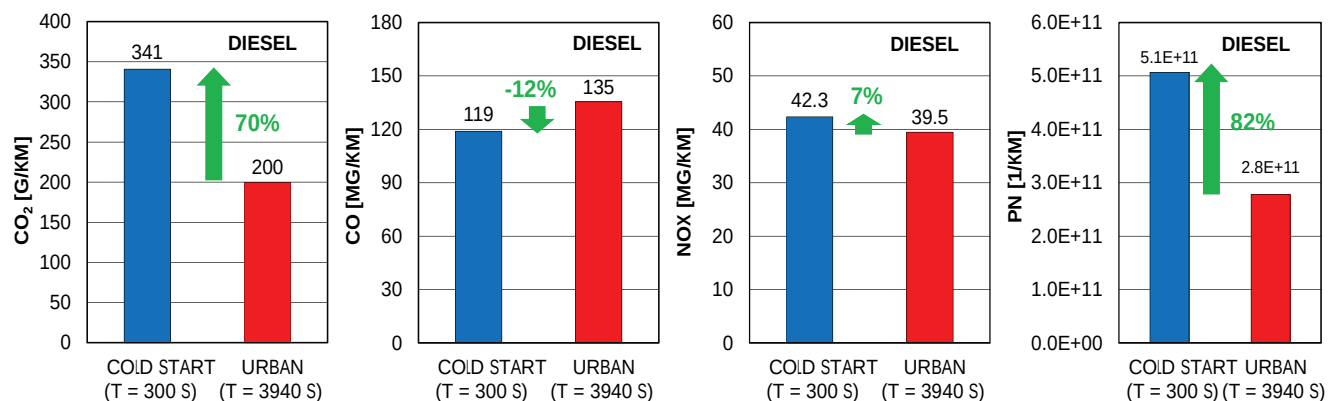


Fig. 13. Road emissions of carbon dioxide, carbon monoxide, nitrogen oxides and particles number during cold start of diesel engine

Comparing the cold start phase for different engine types in the urban part of the RDE test and in the entire RDE test it was found that:

- for gasoline engine (Fig. 14):
 - the share of carbon dioxide emissions during the cold start (300 s) in relation to the urban part of the RDE test is 9.8%, and in the relation to the entire RDE test it decreases to 3.4%,
 - the share of carbon monoxide emissions during the cold start (300 s) in relation to the urban part of RDE test is 10.65%, and in the relation to the entire RDE test it decreases to 4.63%,
 - the share of nitrogen oxides emission during the cold start (300 s) in relation to the urban part of RDE test

is 12.26%, and in the relation to the entire RDE test it decreases to 3.42%,

- the share of particles number emissions during the cold start (300 s) in relation to the urban part of RDE test is 18.37%, and in the relation to the entire RDE test it decreases to 0.05%,
- for diesel engine (Fig. 15):
 - the share of carbon dioxide emissions during the cold start (300 s) in relation to the urban part of RDE test is 11.17%, and in the relation to the entire RDE test it decreases to 3.98%,
 - the share of carbon monoxide emissions during the cold start (300 s) in relation to the urban part of RDE test is 5.57%, and in the relation to the entire RDE test it increases to 8.19%,

- the share of nitrogen oxides emission during the cold start (300 s) in relation to the urban part of RDE test is 7.02%, and in the relation to the entire RDE test it decreases to 2.95%,
- the share of particles number emissions during the cold start (300 s) in relation to the urban part of RDE test is 11.94%, and in the relation to the entire RDE test it decreases to 4.55%

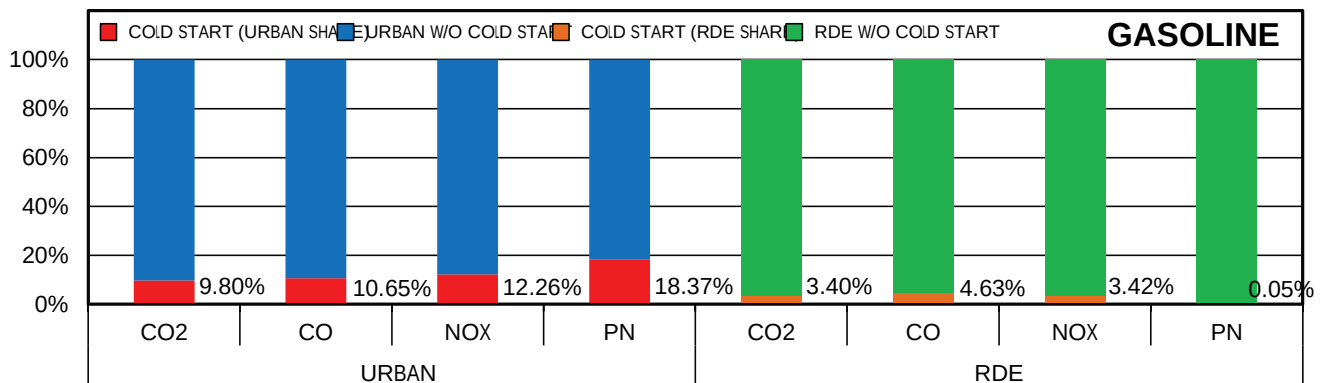


Fig. 14. The share of emissions exhaust compounds during the cold start (300 s) in relation to the urban part of RDE test and the entire RDE test for gasoline engine

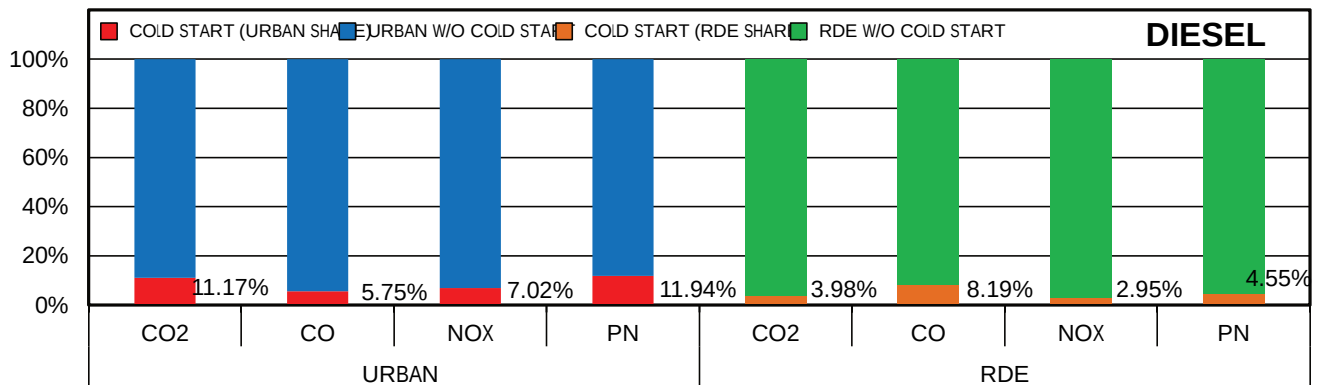


Fig. 15. The share of emissions exhaust compounds during the cold start (300 s) in relation to the urban part of RDE test and the entire RDE test for diesel engine

Summary

Package 3 extended the RDE test procedure by introducing the inclusion of cold start period emissions in the whole value of exhaust emissions. It resulted in an increase of road exhaust emissions. This is mainly due to the warming of exhaust aftertreatment system, and at the same time to the pressure to reduce the exhaust emissions as much as possible in the first period after the cold start.

In the RDE test during cold start phase (300 s period considered) the vehicle with a diesel engine in relation to the gasoline engine is characterized by:

- 2 times lower mass of carbon dioxide (and thus fuel consumption),
- 23 times lower mass of carbon monoxide,
- 16 times higher mass of nitrogen oxides,
- 2 times higher number of particles.

Nomenclature

a	acceleration
a ₊	positive acceleration
b	road exhaust emissions value
CF	conformity factor
EOBD	European on-board diagnostic
EU	European Union
Euro	emission standard
GPS	global positioning system

PEMS	portable emission measurement system
PN	particle number
RDE	real driving emissions
RPA	relative positive acceleration
S	distance
u	share
WLTC	worldwide harmonized light vehicles test cycle
V	velocity

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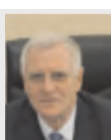
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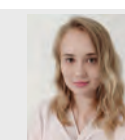
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Analysis of electric motor vehicles market

The increasingly restrictive standards related to exhaust emissions from cars make difficult the development of internal combustion engines. The activities undertaken in the design of internal combustion engines are mainly based on downsizing, e.g. decreasing the engine's displacement. The main direction in the development of vehicle propulsion is to reduce carbon dioxide emissions. It is expected to reduce CO₂ emissions in 2020 to reach 95 g/km. Electric vehicles achieve low noise levels and do not emit a burn, and thus, their use leads to a reduction in the amount of toxic exhaust gases in the air. The aspect of reducing emissions of harmful exhaust compounds and activities focusing on downsizing on the market of combustion engine cars leads to a significant increase the number of electric vehicles. In 2018 around 95 million motor vehicles were registered in the world, of which around 12 million in the European Union and 273 thousand in Poland. The number of electric vehicles among all sold is around 5.5%. Every year new, more technologically advanced models appear on the electric vehicle market. In 2018, the most popular model was the Nissan LEAF and the BAIC EC-Series. A large number of Renault ZOE have also been sold. In article analyzed different models of electric vehicle, which are available on market and presented the characteristics based on e.g. price per 100 kilometers, range for every model or charging time.

Key words: electric propulsion market, toxic exhaust emission, costs of charging electric vehicle

1. Introduction

The increasingly restrictive standards related to exhaust emissions [14, 16] from cars make difficult the development of internal combustion engines. The activities undertaken in the design of internal combustion engines are mainly based on downsizing, which means decreasing the engine's displacement. The main direction in the development of vehicle drives is to reduce carbon dioxide emissions. It is expected that the 2020 CO₂ emission reduction will reach 95 g/km [1]. When analyzing, the market for electric vehicles with electric propulsion, at first the construction of such a propulsion should be known (Fig. 1). The electric drive consists of such elements as: the drive accumulator, engine, tachometer, vehicle network battery, power unit and control unit, converter [3].

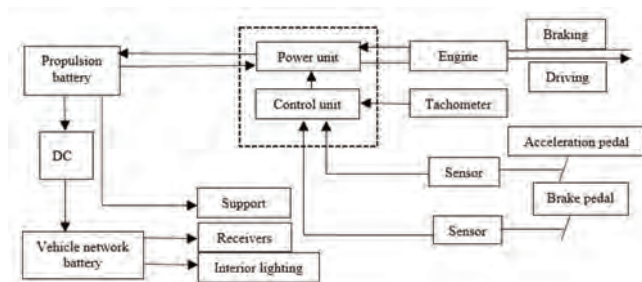


Fig. 1. The scheme of configuration motor vehicle with electric propulsion [3]

The use of electric propulsion lead to achieves low noise levels and do not emit a toxic exhaust emission, and thus, their use leads to a reduction in the amount of toxic exhaust gases in the air. The aspect of reducing emissions of harmful exhaust compounds and activities focusing on downsizing, on the market for combustion cars, leads to a significant increase in the pace of development of electric drives, not only passenger cars, but also various types of aircraft and yachts. In 2018 alone (Fig. 2) 49.1% cars registered in Norway was electric cars, in Iceland it was 19.1%.

The biggest market of electric vehicles are China and USA [2].

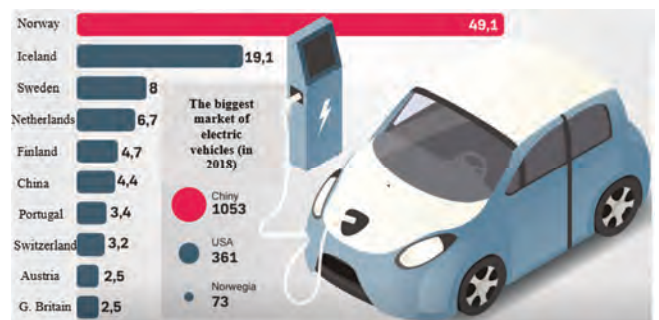


Fig. 2. The part of electric cars in new car registrations in 2018 [2]

The percentage division of fully electric cars among those registered by November in 2018 is shown in Fig. 3.

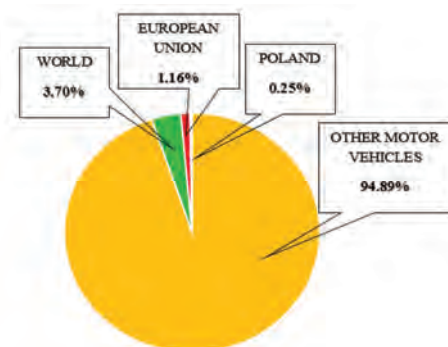


Fig. 3. Percentage part of fully electric cars among those registered by November 2018

Can be seen the number of electric vehicles of all sold cars is around 5.5%. In the European Union, around 1.16% of them were sold, where the largest number of fully electric vehicles were registered in Norway. Every year new, more technologically advanced models appear on the elec-

tric vehicle market. In 2018, the most popular unit was the Nissan LEAF. A large number of Renault ZOE have also been sold. This model has been eagerly bought in Poland, mainly due to belonging to the car sharing system – Traficar.

2. The most modern solutions for electric cars

The latest models of cars with electric propulsion available on the market in 2019 include Hyundai KONA electric (Fig. 4). The Hyundai company was founded in 1967 and is a South Korean automotive group. This model is a Sport Utility Vehicle (SUV) with electric propulsion.



Fig. 4. Hyundai KONA electric [5]

Data regarding the electric drive are shown in Table 1. As can be seen (on the basis of Table 1) Hyundai KONA electric has a very large range and battery capacity with fast charging (up to 80%). In about an hour, charging with a high voltage charger can reach a range of about 300 km. The maximum speed that this car model can reach is about 170 km/h. This vehicle develops a speed of 100 km/h in less than 8 seconds.

Table 1. Technical data for Hyundai KONA electric [5]

Type of data	Value
Battery capacity	64 kWh
Maximum power	204 hp
Maximum velocity	167 km/h
Acceleration (0-100 km/h)	7.6 s
Time of fast charging	54 min
Time of normal charging	9 h 40 min
Range	470 km

One of the most popular electric cars bought in 2018 was Renault ZOE (Fig. 5). It is a car often using in cities, as Traficar (car sharing system in Poland), which is a car rented for a minute's drive. This form of Renault ZOE is from August 2018. The based data concerns this model of car are performed in Table 2.



Fig. 5. Renault ZOE [6]

Analyzing Table 2, it can be concluded that the discussed model is a small unit (4.084 m/1.95 m/1.5 m). It is a car vehicle suitable mainly for moving around the city, but it also provides a range of up to around 300 km (Table 2). The possibility of fast charging ensures a charge up to (80%) in about 2 hours. Renault ZOE is a vehicle mainly used for urban driving. The maximum speed that this model can reach 135 km/h. The disadvantage of this vehicle is that acceleration to 100 km/h takes about 13 seconds.

Table 2. Technical data for Renault ZOE [6]

Type of data	Value
Battery capacity	41 kWh
Maximum power	92 hp
Maximum velocity	135 km/h
Acceleration (0-100 km/h)	13.2 s
Time of fast charging	1 h 40 min
Time of normal charging	7 h 25 min
Range	300 km

When reviewing the market for motor vehicles with fully electric drive, it is necessary to characterize the unit, which in 2018 was the most frequently purchased electric car. In the same year, around 42,000 Nissan LEAF models were sold (Fig. 6).



Fig. 6. Nissan LEAF [7]

Table 3 summarizes the basic technical data for the vehicle. It is a model with a capacity of 40 kWh batteries and a range of 380 km. The obtained power is about 150 hp. Up to 80% can be charged in 60 minutes using a high voltage charger (at least 40 kW). The maximum speed that this vehicle can reach is 140 km/h and up to 100 km/h accelerates in less than 8 seconds.

Table 3. Technical data for Nissan LEAF [7]

Type of data	Value
Battery capacity	40 kWh
Maximum power	150 hp
Maximum velocity	140 km/h
Acceleration (0-100 km/h)	7.9 s
Time of fast charging	60 min
Time of normal charging	6 h 30 min
Range	380 km

In 2018, on the market of fully electric motor vehicles, also e-GOLF occur (Fig. 7). It is one of two models of cars with electric propulsion produced by Volkswagen.



Fig. 7. Volkswagen e-GOLF [8]

The basic technical data on the Volkswagen e-GOLF vehicle drive is shown in table 4. This model has 36 kWh batteries, which deliver 134 hp to the drive. Using the quick-charger chargers most often available in Poland (charging 40 kW), we will charge the car up to 80% in about an hour. The maximum speed that the drive starts is 150 km/h. On the other hand, reaching 100 km/h takes about 10 seconds.

Table 4. Technical data for Volkswagen e-GOLF [8]

Type of data	Value
Battery capacity	36 kWh
Maximum power	134 hp
Maximum velocity	150 km/h
Acceleration (0-100 km/h)	9.6 s
Time of fast charging	60 min
Time of normal charging	6 h
Range	300 km

The next analyzed model is Audi e-TRON (Figure 8). The Audi company is a German car manufacturer belonging to the Volkswagen group. The company was founded in 1909. The premiere of this electric car model in Poland took place in November 2018. This vehicle has a fully electric propulsion and is a SUV. The e-TRON model is the first electric car released by Audi.



Fig. 8. Audi e-TRON [9]

Table 5 contains the technical data. Audi e-TRON has a battery with a very large capacity – 95 kWh. This car with fully charged batteries reaches a range up to 400 km. The obtained power is up to 408 hp. The maximum speed that a vehicle can reach is 200 km/h. The Audi model reaches

speed of 100 km/h in 6 seconds. The disadvantage of the car is the charging time, because the basic charger, available on Polish stations can be charged up to 5 hours.

Table 5. Technical data for Audi e-TRON [9]

Type of data	Value
Battery capacity	95 kWh
Maximum power	408 hp
Maximum velocity	200 km/h
Acceleration (0-100 km/h)	6 s
Time of fast charging	5 h*
Time of normal charging	10 h
Range	400 km
*for charger with 22 kW power	

BMW is a German car company that was founded in 1916. The production includes two models of electric cars: i3 and i8. The i3 model (Fig. 9) will be described. It has been in production since 2013 and belongs to minivan vehicles. According to ACEA (European Automobile Manufacturers Association), in 2017 it was the second most-bought electric car model in Poland [14].



Fig. 9. BMW i3 [10]

Table 6 contains the basic technical data for the BMW i3 drive. The battery capacity is 33 kWh. The car can be charged in about 45 minutes, with a quick charger up to about 80%. The maximum power that can be obtained 170 hp. The vehicle accelerates to 100 km/h in about 7 seconds, and the maximum speed that it can reach is 150 km/h. Batteries allow the range of the car for about 300 km.

Table 6. Technical data for BMW i3 [10]

Type of data	Value
Battery capacity	33 kWh
Maximum power	170 hp
Maximum velocity	150 km/h
Acceleration (0-100 km/h)	7.2 s
Time of fast charging	45 min
Time of normal charging	3 h 45 min
Range	300 km

Jaguar I-PACE S is the first car in this brand's electric car (Fig. 10). The Jaguar company was founded in 1922 and is a British brand owned by the Tata Motors group. The manufacturer's goal was to construct an electric car with high performance. This model is equipped with two electric

motors. The premiere of Jaguar I-PACE was launched in March 2018. Jaguar I-PACE is an SUV.



Fig. 10. I-PACE S Jaguar [11]

Technical data provided by the car manufacturer is shown in Table 7. The battery capacity is very large, it is 90 kWh. Obtained power is 400 hp, and the maximum speed that can reach a car is 200 km/h. The fully charged vehicle provides a range up to 470 km. With a fast charger, the car can be charged up to 80% in about 2.5 hours.

Table 7. Technical data for Jaguar I-PACE [11]

Type of data	Value
Battery capacity	90 kWh
Maximum power	400 hp
Maximum velocity	200 km/h
Acceleration (0-100 km/h)	4.8 s
Time of fast charging	2 h 30 min
Time of normal charging	12 h
Range	470 km

The next discussed model is a Tesla brand car (Fig. 11). This is an American manufacturer of fully electric cars. The company was founded in 2003. Tesla manufactures just 4 models of electric cars. This model – Tesla 3 is the newest and is in production from 2018. It is a medium-class sedan car.



Fig. 11. Tesla 3 [12]

Technical data provided by the manufacturer is included in Table 8. The vehicle has 80 kWh batteries, and the maximum power provided by the electric engine is 257 hp.

When the car is 100% charged, this model has a range of about 500 km. Charging time Tesla 3 with charger with capacity about 40 kW take one and a half hours. The car achieves a top speed of around 250 km/h, while the speed of 100 km/h can develop in about 3 seconds.

Table 8. Technical data for TESLA 3 [12]

Type of data	Value
Battery capacity	80 kWh
Maximum power	257 hp
Maximum velocity	250 km/h
Acceleration (0-100 km/h)	3.3 s
Time of fast charging	1 h 30 min
Time of normal charging	8 h
Range	499 km

3. Comparison of available solutions for electric cars

Analyzing commercially available solutions for fully electric cars, the most important aspects taken into account are the price of the car and of its maintenance, e.g. the cost of one full charge of the car. An important feature of this type of vehicle is also the possible range and speed. In Poland, the largest charging network for electric cars is Greenway. According to data from the Greenway report for 2018, there are currently 115 charging stations in Poland. They are 85–100 km apart (Fig. 12), and the sum of energy is 350 kWh [4].

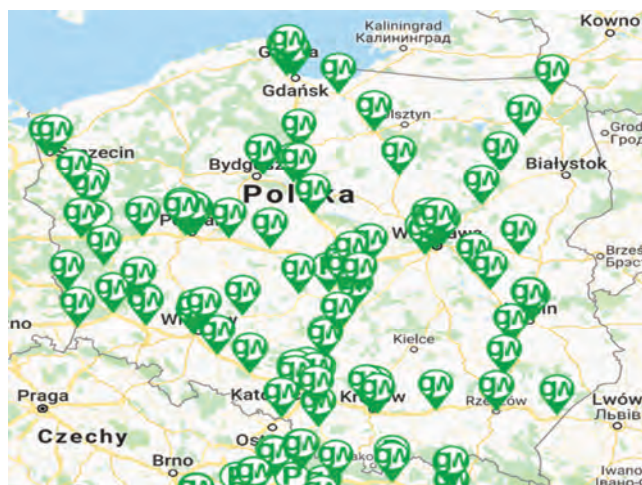


Fig. 12. Location of electric car charging stations in Poland [4]

Currently, chargers in Poland have a charging power of 40 kW, which significantly increases the time of charging the car. The Greenway company plans to introduce 136 new chargers by 2020, mainly fast (50–150 kW) and ultra-fast (350 kW) [4]. These changes will enable charging of electric vehicles in a much shorter time. When analyzing the "cost-effectiveness" and costs associated with having an electric car, should refer to the price list of services provided by Greenway (Fig. 13).

Based on the Greenway price list and technical data for a given electric car model (Table 9), it is possible to calculate the costs of its use.



Pricelist of recharging services in GreenWay network¹⁾
Valid from 01.01.2019

	One-time recharging ²⁾	Multiple recharging ³⁾	
	Energia Ad hoc	Standard plan Energia Standard	Subscription plans Energia Plus Energia Max
Monthly subscription fee	0,00 PLN	0,00 PLN	39,99 PLN 99,99 PLN
DC recharging – nominal power			
Rate for kWh	2,69 PLN	2,19 PLN	1,59 PLN 1,29 PLN
Rate for minute	0,40 PLN	0,40 PLN	0,40 PLN 0,40 PLN
Time after which the payment for minutes is applied	45 minutes	45 minutes	60 minutes 90 minutes
DC recharging – power reduced to 40 kW⁴⁾			
Rate for kWh	2,42 PLN	1,97 PLN	1,43 PLN 1,16 PLN
Rate for minute	0,40 PLN	0,40 PLN	0,40 PLN 0,40 PLN
Time after which the payment for minutes is applied	60 minutes	60 minutes	75 minutes 105 minutes
AC recharging			
Rate for kWh	1,69 PLN	1,39 PLN	1,19 PLN 0,99 PLN
Rate for minute	0,40 PLN	0,40 PLN	0,40 PLN 0,40 PLN
Time after which the payment for minutes is applied	180 minutes	180 minutes	180 minutes 180 minutes
Roaming services⁵⁾			
DC recharging 50 kW for minute			3,30 PLN
DC recharging 22 kW for minute			1,10 PLN
AC recharging 43 kW for minute			3,30 PLN
AC recharging 11-22 kW for minute			1,10 PLN
AC recharging <11 kW for minute			0,30 PLN
Other fees			
Registration fee, including issuing of the first RFID carrier for the driver ⁶⁾	0,00 PLN	0,00 PLN	0,00 PLN
Issuing of the new or additional RFID carrier for the driver ⁷⁾	20,00 PLN	10,00 PLN	0,00 PLN
Recharging initiated by GreenWay operator at the request of the driver ⁸⁾	20,00 PLN	10,00 PLN	0,00 PLN
Additional fee for damage of the infrastructure constituting the GreenWay network or violations listed in the proper terms and conditions of service ⁹⁾	400,00 PLN	400,00 PLN	400,00 PLN

Gross prices, VAT included.
The terms used in the pricelist should be understood in accordance with the GreenWay terms and conditions of services.

1) The total price of the recharging service shall be calculated as sum of (i) subscription fee (ii) fee based on kWh-rate and (iii) fee based on minute rate.
2) Applies to clients using one-time recharging service.
3) Applies to clients who have completed registration in order to use the multiple recharging service.
4) Applies to chargers for which the maximum DC charge power has been reduced to 40 kW. Price for kWh reduced by 10% in comparison to DC recharging – nominal power.
5) Applies to stations owned by GreenWay roaming partners. The power is defined by the maximum available power of the charger. For GreenWay charging stations in the Slovak Republic, the price of the multiple recharging apply.
6) Additional fee is not subject to VAT.

Fig. 13. Price list of Greenway's services [4]

Table 9. Collected data to the analysis [5-12]

Model of electric vehicle	Battery capacity [kWh]	Max. power [HP]	Price [PLN]	Max. velocity [km/h]
Hyundai KONA electric	64	204	136000	167
Renault ZOE	41	92	132000	135
Tesla 3	55	257	255000	250
I- PACE S Jaguar	90	400	356500	200
Nissan LEAF	40	150	155500	140
Volkswagen e-GOLF	36	134	187500	150
BMW i3	33	170	165000	150
Audi e-TRON	95	408	360000	200
Model of electric vehicle	Cost per 100 km	Range (WLTP)	Time of charging (for 7.2 kW charger)	Time of charging (for 40 kW charger)
Hyundai KONA electric	16.2	470	9.7	2 h
Renault ZOE	16.1	300	7.4	65 min
Tesla 3	18.58	310	8.25	1 h 30 min
I- PACE S Jaguar	25.9	470	12.7	2 h 30 min
Nissan LEAF	12.2	380	6.5	60 min
Volkswagen e-GOLF	9.8	300	6	60 min
BMW i3	15.2	300	3.75	45 min
Audi e-TRON	18	400	10	5 h*

Table 9 contains the data necessary to perform the analysis: the amount of kWh consumed per 100 km by each of the car models, their range, battery capacity and charging

time. To perform the characteristics, the maximum speed of the vehicle model, maximum power and price will also be used. Detailed calculations will be carried out for one model – Renault ZOE, and the remaining results compiled in the form of characteristics. The range of the car is the results of tests in accordance with WLTP (Worldwide harmonized Light duty vehicle Test Procedure) [15], which is a cycle of research on CO₂ emission standards, fuel consumption and energy consumption. WLTP tests were introduced in 2017. From that date all cars with the new homologation have values according to WLTP, and from 1 September 2018, all registered vehicles will require WLTP entries [13].

Being a Renault ZOE user (selected for detailed analysis), the costs related to use will be as follows:

- subscription purchase: it is assumed that the user performs full registration and subscribes to Energia standard, which monthly costs 0.00 PLN,
- it is assumed to charge the vehicle with direct current (DC) – with the charger power reduced to 40 kW (such chargers most often occur in Poland).
- Cost of charging vehicle Renault ZOE up to 100%

$$C_{100\%} = P_A \cdot C_{1kWh} + [(T_{char} - 60) \cdot C_{LIMIT}]$$

where: $C_{100\%}$ – cost of charging vehicle up to 100%, P_A – capacity of battery [kWh], C_{1kWh} – cost for 1 kWh charging in energia standard subscription, T_{char} – time of charging with 40 kW voltage charger, C_{LIMIT} – cost of one minute charging after exceed the time limit – in Energia standard subscription after 60 min

$$C_{100\%} = 41 \cdot 1.97 \cdot [(65 - 60) \cdot 0.40] = 82.77 \text{ PLN}$$

- Cost of overpower 100 km by vehicle

$$C_{100 \text{ km}} = W \cdot C_{1kWh}$$

where: $C_{100 \text{ km}}$ – cost for overcome 100 km by each car model, W – number of kWh necessary to overcome 100 km, $C_{1 \text{ kWh}}$ – cost for 1 kWh charging in Energia standard subscription.

$$C_{100 \text{ km}} = 16.1 \cdot 1.97 = 3.72 \text{ PLN}$$

Based on the results, it can be said that users will pay about 83 PLN for driving 300 km (city driving). For comparison, for 300 km, for example Volkswagen Polo, which consumes approximately 7 liters of gasoline for every 100 km, the user will incur costs equal to:

$$C_{VW} = S \cdot W_{l/100km} \cdot C_G = 3 \cdot 7 \cdot 4.8 \approx 100 \text{ PLN}$$

where: C_{VW} – cost of overcome 300 km with Volkswagen Polo vehicle, S – road to overcome, $W_{l/100 \text{ km}}$ – volume of gasoline consumed by vehicle per 100 km, C_G – cost of gasoline.

As can be seen, the result is comparable. The calculations presented were made for all the models discussed, and the results will be presented in the form of characteristics (Figs 14 and 15).

Figure 14 shows the cost of moving 100 km of the electric vehicle models. These costs reach values between 20 and 50 PLN. The lowest cost and at the same time comparable to the price of moving a car with gasoline-powered combustion engine (green color) is characteristic for

Volkswagen e-GOLF and Nissan LEAF cars. In the case of these two models, the price per 100 km is 19.31 PLN and 34.03 PLN respectively. The least profitable types of electric vehicles are SUV cars – Audi e-TRON and Hyundai KONA, for which the cost for covering a distance of 100 km is 35.46 and 31.91 PLN, respectively. The most expensive in operation are cars reaching high values of maximum power and maximum speed – I-PACE Jaguar and Tesla 3 (red color). However, it should be noted, that the price per 100 km for Hyundai KONA electric and Renault ZOE is very similar, although these are two very different types of cars. This means that Hyundai KONA, which is an SUV, is characterized by low costs, in terms of its type.

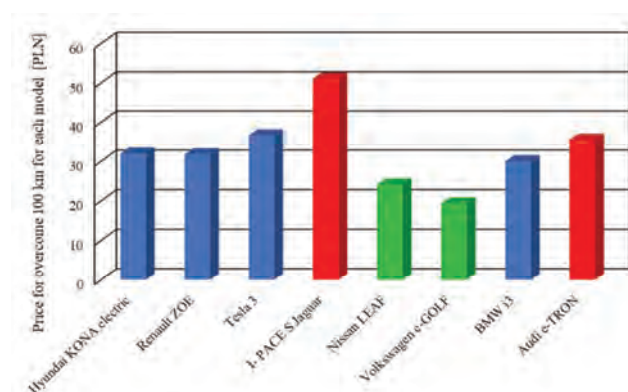


Fig. 14. The price for overcome 100 km for each model

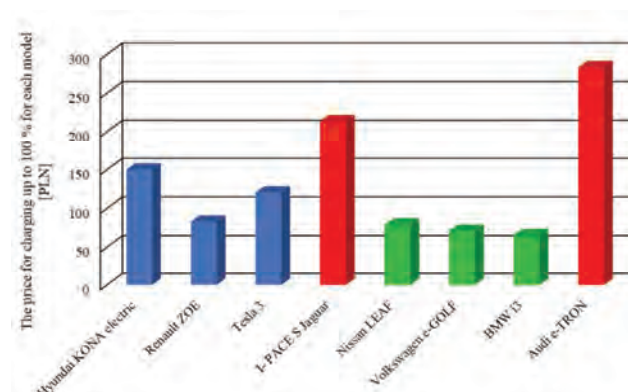


Fig. 15. Price for charging up to 100% for each model

Also, a differentiated charge for up to 100% of selected vehicle types was analyzed (Fig. 15). The lowest price, when loaded up to 100%, is modeled on small passenger cars – Volkswagen e-GOLF, Nissan LEAF and BMW i3 (green color). The range of costs for one charging is from 65 to about 80 PLN. The least favorable results concern I-PACE Jaguar and Audi e-TRON vehicles. In both cases, the price for a full charge differs significantly from the price for the other models and amounts to 213 and 283 PLN. Comparing the two SUV models – Hyundai KONA and Audi e-TRON, it can be seen a significant difference in the cost of use. The Hyundai brand car is about 130 PLN cheaper, despite the similar charging time. This is caused by a significant difference in the capacity of batteries for individual models (64 and 95 kWh). This affects the whole of the obtained results, hence their difference compared to

those obtained during the analysis of the costs of covering a distance of 100 km.

The other characteristics created on the basis of the data from table 9 will be performed to make a more detailed analysis. Figure 16 shows two relations: maximum range (according to WLTP) for individual car models and charging time up to 100% for a 40-kW charger. The largest range is provided by Hyundai, Audi and Jaguar. Can be seen with a long range, these models are characterized by longer charging times.

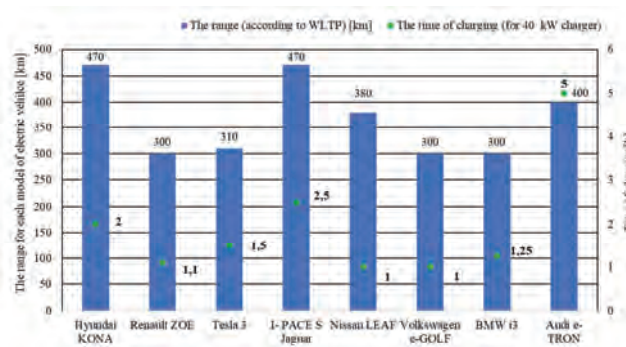


Fig. 16. The range and time of charging for each model

However, the shortest range and the shortest charging times are characterized by the following models: Volkswagen e-GOLF, BMW i3 and Renault ZOE. The most advantageous model at this stage of the analysis is the Nissan LEAF, which, after a full charge, lasting one hour, can provide a range of around 380 km. Tesla 3, on the other hand, despite high maximum power and high performance, can only beat 310 km on a single charge. This may be due to the lower capacity of the battery than in the case of models such as Audi e-TRON, or I-PACE Jaguar, having a battery capacity of about 90 kWh. In the case of Audi, the charging time shown is for a 22 kW charger (no manufacturer's information about charging time for a 40 kW charger). It can therefore be assumed that with a 40 kW charger, this time would be about 2.5 hours or up to a maximum of 3 hours. Based on this assumption, it should be noted that the charging time of the Audi e-TRON model is similar to the other models with a range of around 400/500 km.

The next discussed characteristic is the comparison of the range of individual models and their maximum power depending on the capacity of the batteries. It can be seen, the maximum power depends on the capacity of the accumulators (Fig. 17).

The models of cars with high capacity batteries (in kWh) provide vehicles with high maximum power values. Models such as Audi e-TRON and I PACE Jaguar, which have accumulators with a power of about 90 kWh, obtain maximum power at 400 hp. It should be noted that the Hyundai electric model, despite the greater capacity of batteries than the Tesla 3 car, is characterized by much lower maximum power.

It may be affected by the torque of the engine installed in these models. Smaller vehicles, mainly intended for urban driving (Nissan, Volkswagen or Renault), achieve the

most favorable results in the analysis. They have the lowest prices, their charging, thanks to the smaller capacity of batteries and lower maximum powers, is short, while providing a relatively large range. The Nissan LEAF is by far the best-performing model of an electric car, taking into account all the factors discussed, which also confirms the popularity of this model – in 2018 it was the most-bought fully electric car.

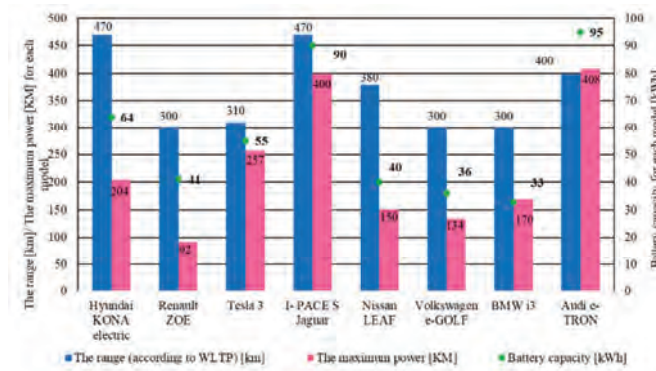


Fig. 17. The range, maximum power and battery capacity for each model

4. Conclusions

The increasingly restrictive standards regarding the emission of toxic exhaust components cause an increase in interest in means of transport with alternative propulsion sources. The article analyzes the available of cars with electric propulsion on the market. The latest and most frequently purchased models of electric cars were selected for

the analysis. When making a comparison in the costs of use and basic technical parameters, no conclusive results were obtained. Referring to the costs: smaller electric cars, serving mainly for city driving, will generate similar costs as cars with internal combustion engines.

However, larger cars—e.g. SUVs generate significantly higher costs of use than cars with an internal combustion engine. Comparing the technical data of individual models, the following conclusions can be made:

- Nissan LEAF, Volkswagen e-GOLF, and Renault ZOE have much worse performance than similar types of vehicles with internal combustion engines,
- Tesla or Jaguar cars are characterized by very good performance, but their use is much more expensive than similar vehicles with internal combustion engines.

In the use of electric cars, the availability of charging points is very important. In Poland, in recent years, many stations have been built, but they are not well equipped. Only a small number of stations have chargers for quick charging. This is important, especially for high-powered cars. Then the charging time using the DC charger is significantly longer. Such situation has a negative impact on the increased sales of electric cars in Poland. To summarize, the availability of fully electric cars on the market is increasing. More and more car manufacturers are choosing to produce cars with fully electric drives. Small city cars, SUVs and sports cars can be found on the market. This shows that electric cars are very popular with the public and greatly affect the development of this type of vehicles.

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Identification of the internal combustion engine malfunctions based on the vibrations of the vehicle body

Selected parameters of an IC engine were correlated with vibrations transferred to the vehicle body with the aim of identifying engine malfunctions. Registered acceleration component signals were analyzed in the aspect of frequency. A reasoning model based on devised identification matrix was used to classify obtained data. This allowed the authors to estimate the possibility of research object damage/malfunctions with an assumed degree of probability.

Key words: vibrations, vehicle diagnostics, Fourier transform

1. Introduction

An internal combustion engine is a machine that transforms chemical energy delivered in the fuel into mechanical work. Due to the fact that in most engines the combustion occurs inside them (within engine cylinders), mechanical work is generated in a repetitive manner. The combustion itself is a very intense process that generates large impulses of thermal energy that is being transferred to the crank and piston system, causing engine vibrations, which are further transferred to the vehicle chassis through the engine mounting points. These vibrations generate a characteristic vibration spectrum during engine operation. The course of this spectrum is dependent on several factors, such as engine type (number of engine strokes), number of cylinders and current crankshaft rotational speed. Characteristic spectral lines can be observed on the vibration spectrum, which are total multiples of crankshaft speed and number of engine cylinders, while also being dependent on engine type. In a single-cylinder four-stroke engine, the combustion process occurs at every other complete turn of the crankshaft, therefore the basic spectral line is equal to 1/2 of crankshaft speed. In nomenclature related to spectral analysis, these are often called the vibrations of the n-th order- in this case of 1/2th order. If a multi-cylinder engine is in good operating condition (meaning the combustion occurs in all cylinders in a similar manner), the vibrations of the n-th order may be very small (from 0.003 to 0.009 m/s), making spectral analysis difficult. In a situation when of the cylinders generates less work than the others (engine malfunction), spectral lines of the n-th order are more pronounced. The signal can then rise above the level of 0.009 m/s to above 0.03 m/s during breaks that occur in the combustion of fuel-air mixture. Several engine malfunctions can contribute to breaks in combustion, including damaged spark plug wires, faulty injector, broken piston ring, low compression ratio or leaking valves. Each of those malfunctions can result in different courses of spectral lines.

IC engines work in a repetitive manner, which means that the dynamic gas forces acting on each of the pistons cause periodic accelerations of the engine crankshaft, which is additionally subjected to torsional, bending and longitudinal vibrations [10]. Therefore, a dynamic gas force acting on each crank of the crankshaft changes periodically with

time, and therefore can be divided into Fourier series. Then, separate elements of the series cause changes in amplitude values for harmonic components of the spectrum [7, 8]. The analysis of vibration velocity peak value of the investigated spectrum is a diagnostic method that is commonly used to assess the machine's technical condition [1–4]. Therefore, spectral components have to be analyzed to locate the malfunction, identify and assess its causes [9]. According to [5, 6], a frequency band of up to n basic ordinates can point to the occurrence of malfunctions in IC engine operation. Basing on that conclusion, an indirect method for diagnosing IC engine malfunctions using the analysis of harmonic components of a vehicle chassis amplitude spectrum.

2. Research object

A Polaris ATV (All-Terrain Vehicle) was used as a research object in this study. It was equipped with an author-devised integrated measurement system comprised of a wireless data transmission system compatible with CAN BUS and an acceleration sensor. The wireless transmitter is plugged into the DTC socket located in the vehicle's engine compartment. After plugging the interface in and turning on the vehicle's ignition, the interface is synchronized with the vehicle on-board data transmission network. The microprocessor built within the interface is connected to a transceiver which converts the signal from the vehicle and sends the data to a computer with the use of a radio transmitter working in the frequency of 433 MHz utilizing UART serial transmission in the range of up to 1 kilometer. The interface is shown in Fig. 1.



Fig. 1. Interface with a USB receiver

The research object was also equipped with a sensor for direct acceleration measurement (3DM-GX3-25). This sensor is made with the use of MEMS technology and is therefore resistant to electrical noise of the conditioning units thanks to using a series of sensors with a PWM impulse output. It has a built-in processor which ensures static and dynamic orientation of measurement axes thanks to utilizing a measurement synthesis algorithm.

Basic parameters of the sensor are given below:

Attitude heading range:	360° about all 3 axes
Accelerometer range:	±5 g standard
Gyroscope range:	±300°/sec standard
Static accuracy:	±0.5° pitch, roll, heading typical for static test conditions
Data output rate:	up to 1000 Hz
Filtering:	sensors sampled at 30 kHz, digitally filtered (user adjustable) and scaled into physical units; coning and sculling integrals computed at 1 kHz
Interface:	RS232
Baud rate:	115,200 bps to 921,600 bps
Power supply voltage:	+3.2 to +16 volts DC

This sensor was used to register vibrations caused by an operating IC engine mounted in the central part of the vehicle chassis (Fig. 2).



Fig. 2. Placement of the 3DM-GX3-25 sensor on the vehicle during tests

The orientation of the sensor in relation to engine crankshaft is as follows: x axis – transverse, y axis – longitudinal, z axis – vertical.

A part of the designed system is a proprietary software solution (Fig. 3) allowing the user to preview current vehicle operating parameters and save the data to the computer's hard disk.



Fig. 3. Author-developed software in operation

2. Signal analysis

Signal registered with the acceleration sensor was analyzed using the Short-Time Fourier Transform (STFT). When using the STFT method, the input signal is first divided into smaller segments, for which the signal can be treated as quasi-stationary. Then, separate segments are processed with the use of FFT algorithm. STFT is described by equation (1), whereas the entry signal window is described by equation (2).

$$\text{STFT}[x_w(t, \tau)] = \int_{-\infty}^{\infty} w(t, \tau) x(t) e^{-j2\pi f t} dt \quad (1)$$

$$x_w(t, \tau) = w(t, \tau) x(t) \quad (2)$$

A comparison of obtained spectra represents a frequency-spectral map of the analyzed signal. A weighted function $w(t, \tau)$ extracts the data segments from the input signal for subsequent FFT analysis.

The results of a Short-Time Fourier Transform (STFT) are obtained as a discrete matrix H , containing a k number of short-time spectra, with a particular number of spectral lines. The number of short-time spectra k (Equation 3) is equal to the number of STFT iterations and is dependent of total input signal sample quantity M subjected to STFT analysis, data segment size N correspondent to time sequence length used for STFT analysis and a time displacement increment for the segment (s).

$$k = \text{Ent}\left(\frac{M-N}{s}\right) + 1 \quad (3)$$

A data segment N should contain a minimum of two signal intervals (important signal component of the lowest frequency). Taking these considerations into account, an optimum number of samples in a data segment for STFT analysis can be calculated using Equation (4).

$$N \geq \frac{2 \cdot f_s}{f_1} \quad (4)$$

Changes in amplitude spectrum obtained with the use of STFT method are shown in Figs 4–6. Obtained courses show the occurrence of clearly visible amplitudes in the basic frequency range f_1 , as well as the other harmonics of

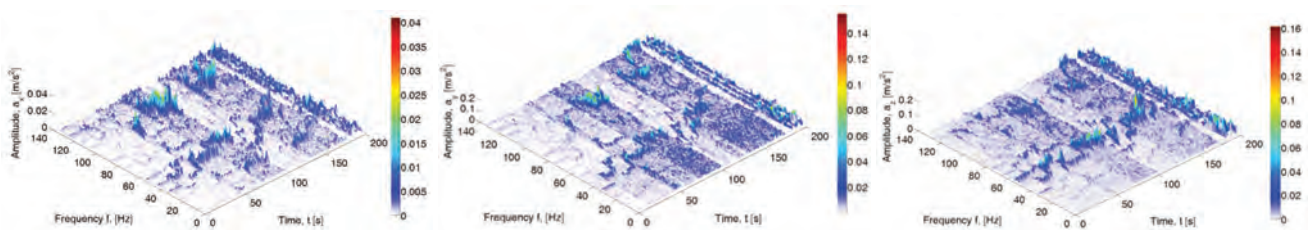


Fig. 4. STFT spectrum for an operational IC engine

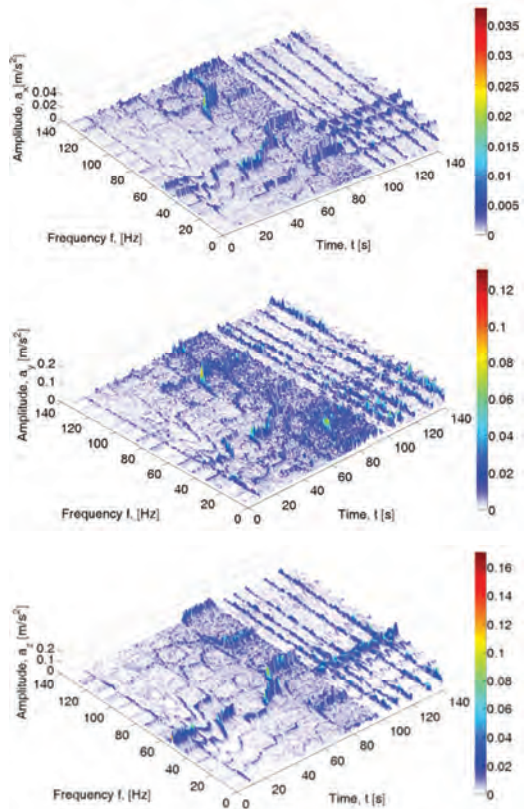


Fig. 5. STFT spectrum for a malfunctioning IC engine – time gap in fuel injection

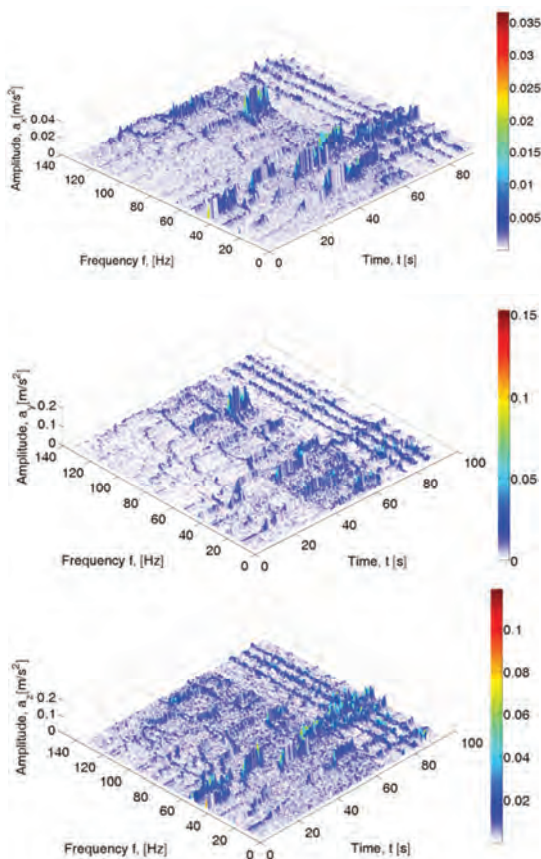


Fig. 6. STFT spectrum for a malfunctioning IC engine – identified air leak in one of the cylinders

the n -th order ($f_{0.5}$, $f_{1.5}$, f_2 , $f_{2.5}$) for different states of the research object. Next, identified engine malfunctions were subjected to the analysis of changes in amplitude spectrum.

An engine rotational speed profile with a time window set to 3 seconds was used to analyze the acceleration signal in the aspect of its frequency, as shown in Fig. 7.

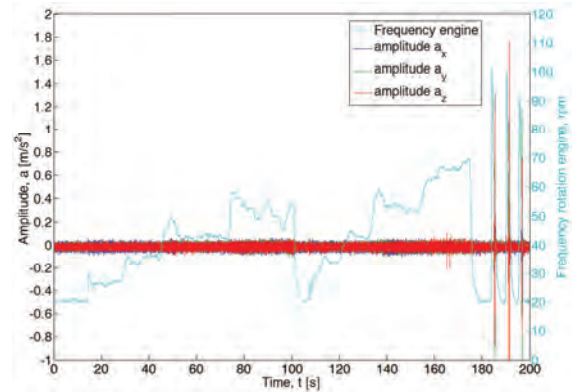


Fig. 7. Engine speed profile for an operational engine

The following acceleration component values were noted for a roadworthy engine operating on idle: $a_x = 0.004$ m/s² for f_1 harmonic, $a_y = 0.03$ m/s² for $f_{1.5}$ harmonic and $a_z = 0.04$ m/s² for $f_{2.5}$ harmonic (Fig. 8). Introducing engine malfunctions in the form of removing an injector or a spark plug in one of the cylinders have resulted in a significant increase of a_x and a_z amplitude values (Fig. 8).

An increase in amplitude values is also visible for other ranges of engine speeds. An example graph showcasing the course for 2500 rpm is shown in Fig. 9.

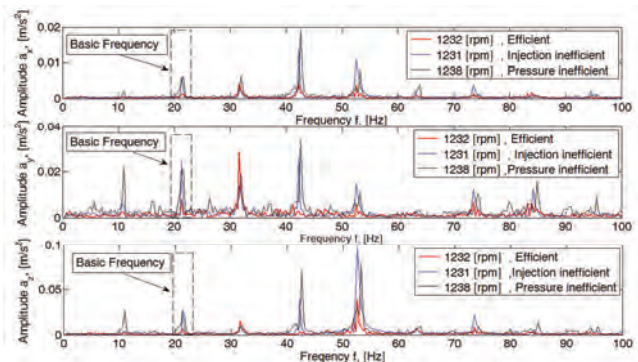


Fig. 8. FFT spectral amplitude for 1200 rpm

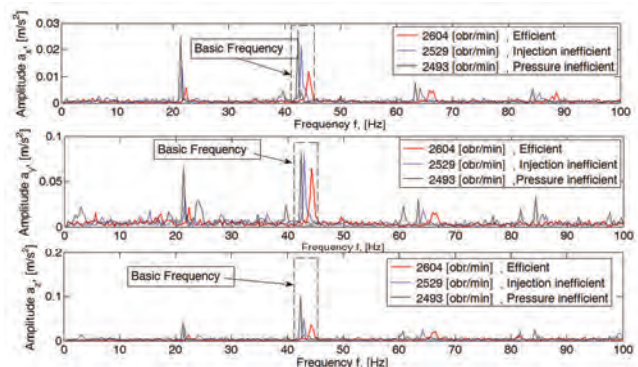


Fig. 9. FFT spectral amplitude for 2500 rpm

Basing on the known crankshaft speed, its base frequency f_1 was determined with the use of ± 1.5 Hz frequency intervals. Next, dominant values of acceleration components of the n -th order ($n = f_{0.5}, f_1, f_{1.5}, f_2, f_{2.5}$). These values were labelled as $a_{x0.5}, a_{x1}, a_{x1.5}, a_{xn}$ and entered into the input data table (Table 1). Determined acceleration components correspond to a particular measurement axis (i.e. a_x) and engine speed (i.e. 1200 rpm). Input data tables were formulated for corresponding measurement axes and engine speeds. Next, relative values for different harmonics in relation to base frequency were devised. Obtained values were labelled A_x for corresponding harmonic components of the signal spectrum.

Table 1. Input data table

f	a_x	Relative value	A_x
$f_{0.5}$	$a_{x0.5}$	$\frac{a_{x0.5} * 100}{a_{x1}}$	$A_{x0.5}$
f_1	a_{x1}	$\frac{a_{x1} * 100}{a_{x1}}$	A_{x1}
$f_{1.5}$	$a_{x1.5}$	$\frac{a_{xn} * 100}{a_{x1}}$	A_{xn}
f_2	a_{x2}		
$f_{2.5}$	$a_{x2.5}$		

Terms specifying relations for harmonics of the spectrum were assumed for residuals. Obtained values are relative to signal values of the base frequency (Table 2).

Table 2. Residuals specifying table for 1200 rpm

No resid.	Limit state WS	Logical value MD of benchmark data for specified engine condition		
		Operational	No injector	No spark plug
1	$A_{x1.0} > A_{x0.5}$	MD11 = 1	MD21 = 1	MD31 = 1
2	$A_{x1.5} > A_{x1}$	MD12 = 1	MD22 = 1	MD32 = 0
3	$(A_{x1.5} + A_{x0.5}) > A_{x1}$	MD13 = 1	MD23 = 0	MD33 = 0
4	$A_{x2} > A_{x1.5}$	MD14 = 1	MD24 = 1	MD34 = 1
5	$A_{x2.5} > A_{x2}$	MD15 = 1	MD25 = 0	MD35 = 1
6	$(A_{x2.5} + A_{x1.5}) > A_{x2}$	MD16 = 1	MD26 = 1	MD36 = 0

Table 3. Benchmark values of residuals for 1200 rpm

Limit state WS	Logical value ME for experimental data
$A_{x1.0} > A_{x0.5}$	ME1
$A_{x1.5} > A_{x1}$	ME2
$(A_{x1.5} + A_{x0.5}) > A_{x1}$	ME3
$A_{x2} > A_{x1.5}$	ME4
$A_{x2.5} > A_{x2}$	ME5
$(A_{x2.5} + A_{x1.5}) > A_{x2}$	ME6

The data of the reference matrix (MD1n, MD2n, MD3n) of individual residues (from 1 to 6) are then compared with the residues of the analyzed signal Men (Table 3). If the compared data is compatible (eg MD12 and ME2), the value of WI in table 4 is 0.

This solution does not precisely point towards the existing identification state of the object. Instead, it shows the probability of its symptoms occurring in relation to the research object. The biggest value of MD-n sum of determinants identifies the occurrence of a particular malfunction with specified probability.

For the identified engine state, a set of chosen possible malfunctions WI was determined. These malfunctions are understood as destructive occurrences leading to deterioration in engine operation.

Table 4. IC engine identification table for 1200 rpm

Comparison of readings (logical differential)	Logical value of state identification determinant WI		
	0 = compatible 1 = incompatible	0 = compatible 1 = incompatible	0 = compatible 1 = incompatible
MD11 $\Leftrightarrow$ ME1	0 = compatible 1 = incompatible	0 = compatible 1 = incompatible	0 = compatible 1 = incompatible
MD12 $\Leftrightarrow$ ME2			
MD13 $\Leftrightarrow$ ME3			
MD14 $\Leftrightarrow$ ME4			
MD15 $\Leftrightarrow$ ME5			
MD16 $\Leftrightarrow$ ME6			
	Percentage share WS1n = 0	Percentage share WS3n = 0	Percentage share WS3n = 0

A diagnostic matrix-based reasoning algorithm was created to identify engine malfunctions. The diagnostic signal should allow the user to detect faults which cause the appearance of state identification determinant, which assumes logical values of 1 or 0 (symptom). RFS relation for the investigated signal can be defined with the use of Binary Method.

3. Identification matrix

Obtained experimental data was analyzed with the use of described method basing on diagnostic matrix to identify malfunctions of the research object. A working example was shown for experimental data obtained for an engine speed of 1200 rpm. Values of acceleration components for x, y, z axes, specific frequency ranges (in relation to base frequency) and assumed state of the engine were shown in Fig. 8. Relative values were then determined for this portion of experimental data.

Table 5. Data analysis for 1200 rpm in x axis

No resid.	MDx 1	MDx 2	MDx 3	ME x			
					WIx1	WIx2	WIx3
1	1	1	1	1	0	0	0
2	1	0	1	0	1	0	1
3	1	0	1	0	1	0	1
4	1	1	1	1	0	0	0
5	0	0	0	0	0	0	0
6	1	0	0	0	1	0	0
WSx [%]					50	100	67

Benchmark data for diagnostic matrices MDx, Mdy, MDz (Tables 5–10) was obtained basing on the analysis of acceleration components and engine condition.

The determinants of the relationship (residue) of particular harmonic axes x, y, z for the efficient engine were marked in columns: MDx1, Mdy1, MDz1. For engine with damaged injection MDx2, Mdy2, MDz2. For a motor with

a non-sealed combustion chamber: MDx3, MDx3, MDx3. The markers of the analyzed signal were denoted as: MEx, MEy, MEz.

Table 6. Data analysis for 1200 rpm in y axis

No re-sid.	MDy1	MDy2	MDy3	MEy	Wly1	Wly2	Wly3
1	1	1	0	1	0	0	1
2	1	0	0	0	1	0	0
3	1	0	1	0	1	0	1
4	0	1	1	1	1	0	0
5	1	0	0	0	1	0	0
6	1	1	0	1	0	0	1
WSy [%]					33	100	50

Table 7. Data analysis for 1200 rpm in z axis

No re-sid.	MDz1	MDz2	MDz3	MEz	Wlz1	Wlz2	Wlz3
1	1	1	0	1	0	0	1
2	1	0	0	0	1	0	0
3	1	0	1	0	1	0	1
4	0	1	1	1	1	0	0
5	1	1	1	1	0	0	0
6	1	1	1	1	0	0	0
WSz [%]					50	100	67

Table 8. Data analysis for 2500 rpm in x axis

No re-sid.	MDx1	MDx2	MDx3	MEx	Wlx1	Wlx2	Wlx3
1	1	1	0	0	1	1	0
2	1	1	1	1	0	0	0
3	1	1	1	1	0	0	0
4	0	0	1	1	1	1	0
5	1	0	0	0	1	0	0
6	1	1	1	1	0	0	0
WSx [%]					50	67	100

The individual MDn reference data were compared with residues of the tested Men's signal, obtaining information on existing relations. In the case of compliance (eg MDx1 (1) and MEx (1)) for 1200 rpm (Table 5), the value of Wlx1 (1) is 0. In the case of non-compliance (eg MDx1 (2) and MEx (2)), the value of Wlx1 (2) is 1.

By analyzing specific measurement data for the engine, a percentage probability of its state was obtained. Data analysis for identified malfunctions in the form of time gap in fuel injection and air leak in one of engine cylinders is shown in Tables 5–7. Show case data analysis for 1200 rpm, whereas Tables 8–10 show data analysis for 2500 rpm.

Nomenclature

x(t) time course representing the analyzed input signal
 τ time window position in function of time
F frequency

Identification matrices allow to identify time gap in fuel injection with 100% certainty (in x, y, z axes) and an air leak within the engine cylinder with 67% probability (in x, y, z axes).

Table 9. Data analysis for 2500 rpm in y axis

No re-sid.	MDy1	MDy2	MDy3	MEy	Wly1	Wly2	Wly3
1	1	0	0	0	1	0	0
2	1	1	1	1	0	0	0
3	1	1	1	1	0	0	0
4	0	0	0	0	0	0	0
5	1	1	0	0	1	1	0
6	1	1	1	1	0	0	0
WSy [%]					67	83	100

Table 10. Data analysis for 2500 rpm in z axis

No re-sid.	MDz1	MDz2	MDz3	MEz	Wlz1	Wlz2	Wlz3
1	1	0	1	1	0	1	0
2	1	1	0	0	1	1	0
3	1	1	1	1	0	0	0
4	0	1	1	1	1	0	0
5	1	1	0	0	1	1	0
6	1	1	1	1	0	0	0
WSz [%]					50	50	100

As shown by the tests, the master data matrices should be determined for specific engine speed values, which can be seen by comparing the Table 5 (1200 rpm) and Table 8 (2500 rpm).

4. Summary and conclusions

The presented research has shown that it is possible to identify engine malfunctions by analyzing the harmonic components of vehicle chassis vibrations with the use of residuals. Two distinct engine failures were purposefully introduced to an investigated spark-ignition engine mounted in RZR 1000 ATV-time gap in fuel injection and air leak in engine cylinder. Research was conducted for different engine speeds operating in idle conditions.

Differences in spectral lines stemming from engine malfunctions were noted when analyzing relative values of harmonic amplitude components. This allowed the authors to formulate a reasoning matrix for identifying engine failure basing on comparative analysis of harmonic components. In future research, it would seem beneficial to conduct a similar analysis of harmonic components for different engine speeds and to specify a diagnostic criterion.

T time
 f_s input signal sampling frequency

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The gas-air mixing systems for the industrial gas engine fuelled with low quality gas

Modifications of standard fuel system for an industrial gas engine and an analysis of the engine operation on low quality fuel have been described in this paper. The implemented fuel system modifications aim to overcome the challenges which arise when standard fuel gas is replaced by low quality one. Analysis focuses on stability of mixture formation esp. on variations in gas distribution for each cylinder of the engine. The results and discussion include also emissions optimization impact on efficiency as well as the efficiency and emissions level dependency on fuel type.

Key words: industrial gas engine, low quality fuels, genset efficiency, emissions

1. Introduction

As power consumption globally increases and fossil fuels consumption is scheduled to be reduced, there is a need to look for new fuels for power generation. Renewable fuels are getting more and more attention but the availability is still low and price is high. The worldwide power generation industry turns its eyes towards wastes and the methods for their effective use for power generation.

The industrial waste and biowaste can be turned into gas or liquid fuel. The technologies are already well known and in commercial use for many years. The main challenge is low fuel quality comparing to conventional fossil fuels. Thus, their application using conventional technologies is limited mainly to boiler combined with steam turbine. Such solution is, unfortunately, expensive and inefficient. The industrial gas engine can be considered as interesting alternative, but some challenges need to be solved.

The standard fuel system of the industrial gas engine is designed and optimized for high quality fuel, such as natural gas or biogas. Moreover, the industrial engines require very precise control of mixture composition, to ensure stable operation at desired load profile. Due to this, the gas fuel system needs to be fast in response and should provide proper cylinder balancing.

The important factor is the engine operation optimization which needs to balance two contradicting targets – low emissions and high efficiency. Typical approach is to reach required level of emissions, as an obligation, and the efficiency is a result. However, for low quality waste fuels efficiency is not as important as for good quality ones mainly due to much lower fuel cost. Nevertheless, available fuel energy should have always been used in the most effective way. The engine controls should be set for fuel properties as different fuel type or composition may influence the efficiency and emissions.

2. Medium pressure gas mixer

The technology of medium pressure gas mixers is an attractive compromise between low pressure and high pressure system (employing direct gas injection). It provides relatively high safety due to low volume of combustible mixture in the intake manifold when compared with low pressure system. At the same time it doesn't require enormous

amount of energy for gas compression as in Gas Diesel technology.

There are however, challenges to be overcome. The fuel needs to be well mixed with air before it is directed to individual cylinders. Since the fuel dosing system in that concept is located after a turbocharger, which is usually close to the intake manifold the system should employ features enhancing mixture process.

In this study we present current status of a work-in-progress development project of a medium pressure fuel system. So far two different concepts of fuel supply system were tested. The two designs were evaluated in terms of air excess ratio distribution between the engine cylinders.

The fuel system was aimed to effectively work on extremely low calorific gases such as synthesis gas as well as to provide fuel flexibility and work on biogas and natural gas.

The experiments were performed on a model intake system to simulate engine conditions. The tests were done for three gases and the flow conditions were properly adjusted for natural gas, biogas and synthesis gas to ensure required mixture composition for each fuel. Both fuel system concepts were providing the gas just after the charge air cooler in a single point in the center of the intake duct. The only difference between the two fuel system concepts was an additional static mixing element referred to as static mixer, which was only applied in a second design.

The uniformity of gas distribution between the cylinders was evaluated by means of normalized equivalence ratio which is an equivalence ratio (ϕ) of the gas supplied to a certain cylinder normalized by an average equivalence ratio (ϕ_{average}) for whole engine. The system performance for natural gas flow settings is shown in Fig. 1.

As shown in Fig. 1 the maximum deviation of normalized equivalence ratio from the average for the base system design (without static mixer) was only 3%. This value was recorded for cylinder number 4. The additional static mixing element didn't improve the gas distribution. For the design with static mixer the maximum deviation of normalized equivalence ratio has increased up to 8% (for cylinder number 3).

When the flow settings were changed to simulate flow rates of biogas the results have changed as shown in Fig. 2.

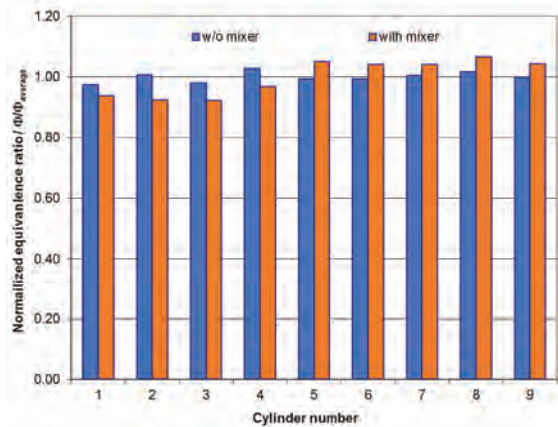


Fig. 1. Comparison of normalized equivalence ratio of two fuel system concepts for natural gas relevant flow conditions

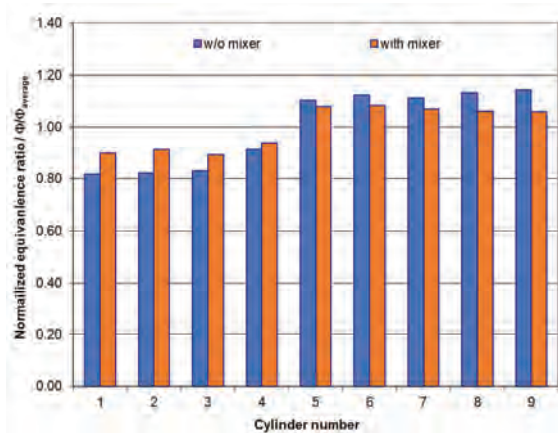


Fig. 2. Comparison of normalized equivalence ratio of two fuel system concepts for biogas relevant flow conditions

The system performance for biogas flow settings has dramatically decreased. The differences in equivalence ratio suggest that most of the gas tends to travel towards one side of the intake receiver. Moreover, in contrary to previous case (natural gas flow settings) in this case the additional static mixing element improved the distribution of the gas.

When the flow settings were changed to simulate mass flow rates of syngas the results have changed again. The results for syngas mass flow settings are presented in Fig. 3.

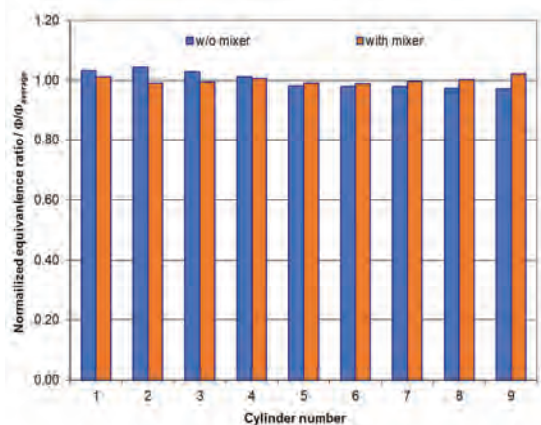


Fig. 3. Comparison of normalized equivalence ratio of two fuel system concepts for syngas relevant flow conditions

For syngas flow settings system performance has been significantly improved. The static mixer has decreased the differences in lambda between the cylinders even more. The system with the additional static mixing element improved the distribution of the gas so the maximum deviation from average was only 2% for cylinder number 9.

The obtained results confirmed that proper mixing in medium pressure systems poses a challenge. Although the fuel system design with static mixer provided sufficient performance for syngas flow settings it appeared to be sensitive to the flow conditions, and on other settings the performance has dramatically dropped.

As explained earlier this is a work-in-progress development project and basing on acquired results the next designs will be created.

3. Low pressure gas mixer

An advanced gas supply and engine control system HE-MUZG can be combined with medium pressure gas fuel injectors or low pressure gas fuel mixer [1–3]. The system allows continuous control of the air and gas fuel flows and automatic adjustment of each one dependent on engine operation conditions and gas fuel quality. The low pressure gas fuel supply point is located in typical way, i.e. in the engine intake air stream, right after air filter and before inlet to turbocharger compressor.

The location of the gas fuel supply point creates a long distance available for mixing but on the other hand the mixture composition is created globally and cannot be adjusted for each cylinder individually. The final gas fuel delivery to each cylinder was measured indirectly by monitoring the combustion process in each cylinder, for example by measuring the exhaust gas temperature, the knock margin, the spark ignition energy etc. The achieved results for individual cylinders balancing, referred to average, are presented in the following Figs 4 and 5.

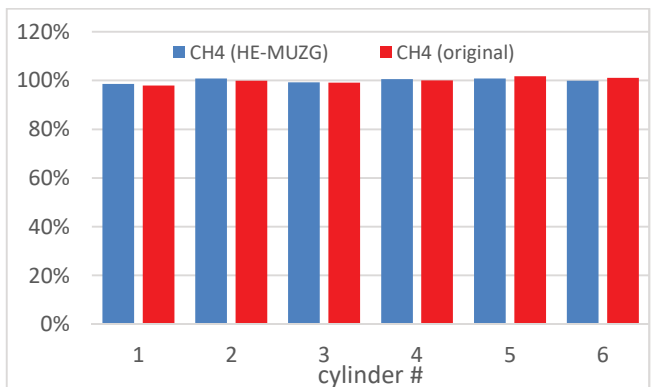


Fig. 4. Individual cylinders balancing comparison based on combustion monitoring for original mixer and HE-MUZG system (balancing referred to average)

The Figure 4 shows that HE-MUZG system provides mixing and cylinder balancing on the same level as standard gas fuel mixer dedicated for methane. The next analysis was done for several fuel types, such as pure propane C_3H_8 , propene C_3H_6 and carbon monoxide CO , biogas (mix of methane CH_4 and carbon dioxide CO_2) and nitrogen-rich natural gas (mix of methane CH_4 and nitrogen N_2).

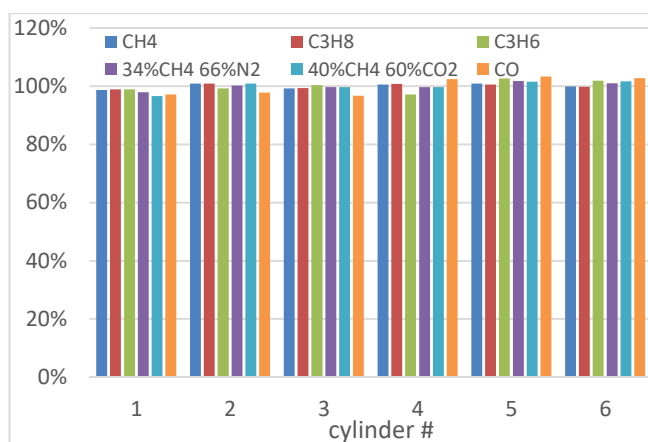


Fig. 5. Individual cylinders balancing comparison for various fuels and HE-MUZG system (balancing referred to average)

The best cylinder balancing was achieved for methane and propane and the worst results were for propene and carbon monoxide. Additional analysis showed that for these two last fuels combustion process is very sensitive and strongly dependent on mixture composition. The carbon monoxide combustion reactions are very slow and even small change in mixture composition can speed up or slow down combustion significantly. On the other hand, the propene is very keen to lead into knock combustion and a small change of mixture stoichiometric coefficient can strongly influence the knock margin. Other observation was that carbon dioxide mixes slightly worse than nitrogen, which is a result of much different molecular weight and density of each gas.

4. Emissions and efficiency

The emissions and efficiency test were focused on NO_x emissions, as this parameter is currently the most challenging to meet. The genset emissions are regulated inde-

pendently by each country or region but the most common used is German air quality norm called "TA-Luft", which even if not used directly is very often used as reference. For example: TA-Luft allows NO_x emission on the level of 500 mg/m³ of the exhaust gas with 5% oxygen content and 1/2TA-Luft allows only half of it [4]. The genset performance tests were carried out for four cases – NO_x emission according to TA-Luft norm, 1/2TA-Luft, maximum efficiency and the lowest possible emissions enabling stable genset operation. The tests were performed for pure methane, biogas (45% CH₄ + 55% CO₂), methane and propane mix (75% CH₄ + 25% C₃H₈), and carbon; with two load levels: 50% and 100% of nominal genset load. The test results are presented into Figures 6 to 9.

The carbon monoxide tests were very challenging and it was possible to measure emissions and efficiency up to 50% genset load and only for optimal control settings, i.e. maximum efficiency.

In general the best performance results were achieved for pure methane. The reason is that the engine was originally configured for methane operation and HE-MUZG system influences only controls and fuel supply leaving the engine hardware unchanged.

The well-known relations between efficiency and NO_x emissions is clearly visible – the lower emissions the lower efficiency. Recently introduced change of allowed emission level from TA-Luft to 1/2TA-Luft leads to increase of gas fuel consumption at nominal genset load by 2.7% for methane 7.9% for biogas and 13.8% for methane mixed with propane. Additionally the change from the optimum genset performance to the lowest possible emission results in gas fuel consumption increase by 10.6% for methane, 12.4% for biogas and 31.2% for methane mixed with propane (all for nominal genset load). This reflects in corresponding increase of annual fuel cost.

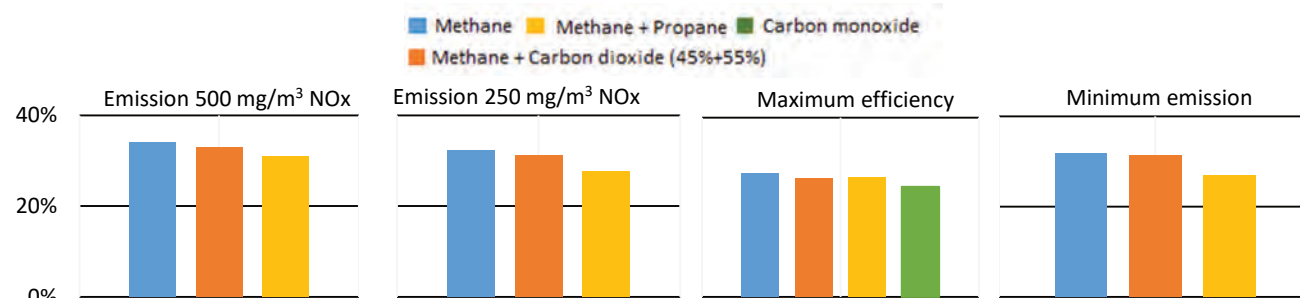


Fig. 6. Genset efficiency at 50% load for different NO_x emission settings

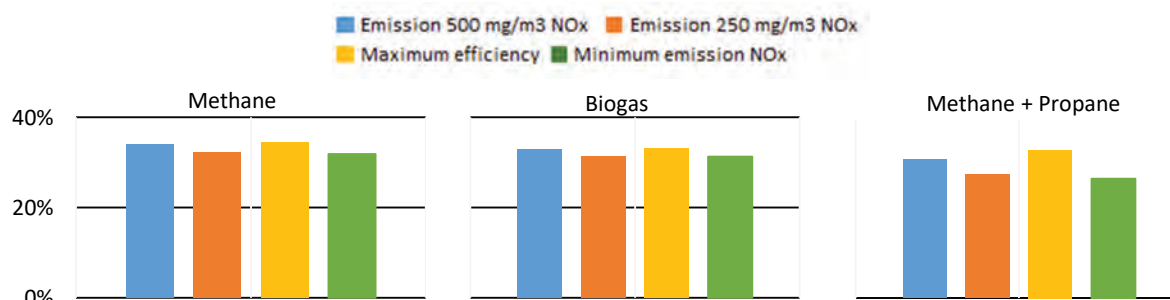


Fig. 7. Genset efficiency at 50% load for different fuels

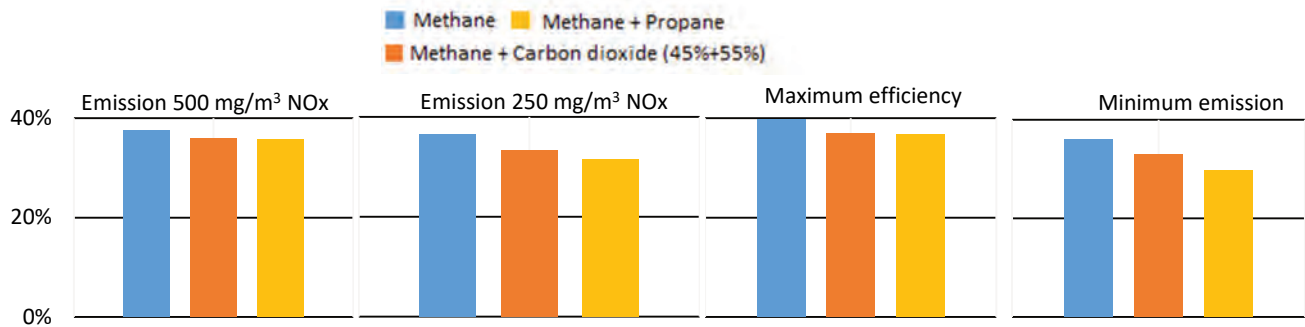
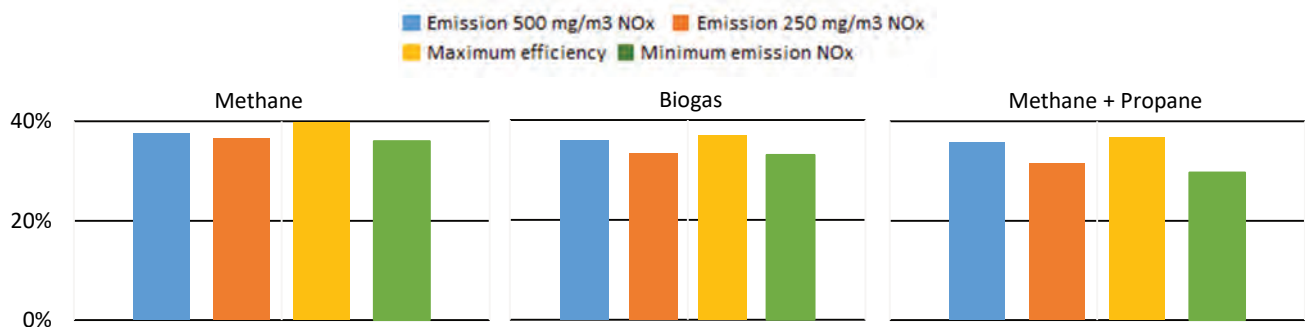

Fig. 8. Genset efficiency at 100% load for different NO_x emission settings


Fig. 9. Genset efficiency at 100% load for different fuels

5. Summary and conclusions

The gas fuel mixing and cylinder balancing results show that much better results are achieved with low-pressure mixer. The main reason is that this solution provides long distance for mixing and also the air and gas passing turbo-charger's compressor get additional mixing enforcement. However, it is important to note that medium pressure gas mixer provides much better mixing for syngas than for natural gas. The influence of fuel type is also visible in low

pressure gas mixer but the change is not that significant as in medium pressure gas mixer.

The emissions measurements showed two important factors. First of all, the fuel type and related to it combustion process strongly influences the NO_x emissions and efficiency. The second factor is that NO_x optimisation enforced by legislation can lead to significant increase of power generation sets operational cost due to huge increase in gas fuel consumption.

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Comparison of fuel consumption and exhaust emissions in WLTP and NEDC procedures

Fuel consumption achieved in the New European Driving Cycle (NEDC) could be 50% lower than the fuel consumption in real driving conditions and in the case of emissions of regulated toxic compounds the differences could even be much greater. In order to bring the results achieved in official tests closer to real life figures, the European Commission introduced in 2017 the Worldwide Harmonized Light Vehicles Test Procedure (WLTP), which replaced the NEDC. In this article the results of fuel consumption and exhaust emissions for 3 cars fitted with engines of the same displacement but with direct and indirect gasoline injection, determined according to the NEDC and WLTC were presented. The results show that the effect of driving cycle on the fuel consumption is equivocal – for one car, fuel consumption was higher in the WLTC; for the other one in the NEDC; and for the third one, fuel consumption achieved in both driving cycles was practically the same. Emissions of regulated exhaust compounds, except for THC, obtained in the WLTC were higher than in the NEDC driving cycle.

Key words: exhaust emissions, fuel consumption, driving cycle, type-approval, chassis dynamometer

1. Introduction

Determination of fuel consumption and emissions of toxic compounds of passenger vehicles is important from points of view of environment protection and prospective car buyers. Fuel consumption achieved in the New European Driving Cycle (NEDC) sometimes could be up to 50% lower than the fuel consumption in real driving conditions, and in the case of emissions of regulated toxic compounds the differences could be even much greater. It is estimated that on average the divergence between type-approval based on the NEDC and real-world fuel consumption of new passenger cars in Europe was increasing over time, from 8% in 2001 to 40% in 2014 [6]. On road emission of NO_x was estimated to be on average 6–7 times higher than the values measured in the NEDC [9].

The divergence of type-approval and real-world values varied by segment, manufacturer, fuel, transmission and so on. Generally that divergence was greater for passenger vehicles with diesel engines than with spark ignition (SI) engines [1, 4, 6].

The reason for the above discrepancies were different conditions of vehicle operation during official testing in the laboratory according to the NEDC and during real vehicle use on the road. Those different conditions resulted of different driving cycle (speed profile, gear shift points) and vehicle configuration (mass, equipment, pressure and type of tires and so on).

In order to bring closer the results achieved in official tests to real life figures, the European Commission introduced in 2017 the Worldwide Harmonized Light Vehicles Test Procedure (WLTP), which replaced the NEDC. The new procedure introduces a more dynamic World Harmonized Light Duty Vehicle Test Cycle (WLTC) and other changes in testing conditions, including various vehicle configurations.

The effect of replacing the NEDC with the WLTP was evaluated in many studies. The CO₂ emission is on average 11% higher for the WLTP worst case scenario (highest vehicle mass and highest rolling resistance), and 1% higher

for the WLTP best case scenario (smallest vehicle mass and rolling resistance), in comparison to the emissions achieved in the NEDC [5]. Results from this study also show that moving from NEDC to WLTP has a stronger impact on diesel than on gasoline vehicles.

On the other hand, the estimations presented in [2] indicate that fuel consumption and CO₂ emission in WLTP tests are on average 6% lower than in NEDC tests (in the case of SI engines – 10%). NO_x emission is 1% higher in WLTP cycle, comparing with NEDC, however for SI vehicles only, it is lower by 16%. Estimated HC emission for gasoline vehicles is 26% lower in WLTP than in NEDC.

Simulations presented in [7] indicate that the CO₂ emission in WLTP will be by 20 g/km greater than in NEDC for the fleet of passenger cars. According to [3] this increase will be equal to about 26 g/km, but the gap between real-world and WLTP CO₂ emissions will still remain, although will be smaller. These statements are only partially confirmed by the results presented in [8]. In the case of vehicles with diesel engines CO₂ emissions achieved in RDE (real driving emission) tests were by 6% higher than achieved in WLTP tests. However, in the case of vehicles with SI engines the CO₂ emissions were 2% lower. NO_x emissions in RDE tests were higher by 20% for vehicles with diesel engine, and by 6% for vehicles with SI engines. At the same time, the authors emphasize the great impact of testing conditions (cold operation, road grade, trip selection and driving style) on results achieved in RDE tests.

Summarizing, the influence of the new testing procedure (WLTP) on the emissions is not obvious, especially in the case of vehicles with SI engines. To evaluate the effect of the driving cycle on fuel consumption and exhaust emissions, 3 cars fitted with SI engines were tested using the NEDC and WLPC cycles. It should be underlined that driving cycle is a relevant part of the test procedure, but it is not the only important factor. The method and parameters used for simulation of vehicle mass and road load are pretty important. The ambient conditions, i.e. temperature, also exert an influence on the results. This paper focuses on

differences caused by the driving cycles themselves, and so the broader test procedures is not examined and discussed in detail.

2. Objects and method

2.1. Vehicles tested

Two of tested vehicles were mini cars (segment A) and one a medium car (segment C). The distances covered by the cars before the tests were different: one of them was almost new, just well run-in (2000 km), the second had mileage of 20000 km and the third of 60000 km. All of the cars tested were in a very good technical state. All cars were equipped with manual gearboxes and four-cylinder, spark ignition engines of the same displacement of 1.4 dm³. One of the engines was naturally aspirated and two were turbocharged, and one engine had direct gasoline injection and two indirect multipoint injection system. A detailed specification of the tested vehicles and engines is presented in Table 1.

Table 1. Specification of tested vehicles

Parameter	Vehicle A	Vehicle B	Vehicle C
Vehicle category	M1	M1	M1
Vehicle segment*	C	A	A
Vehicle mass [kg]	1360	1160	1130
Vehicle mileage [km]	60000	20000	2000
Engine type	SI / R4	SI / R4	SI / R4
Swept volume [dm ³]	1.4	1.4	1.4
Injection / charging	DI / turbo	MPI / turbo	MPI / –
Engine power [kW]	90	107	57

*A – mini cars, C – medium cars

2.2. Laboratory

Tests were carried out in BOSMAL's exhaust emissions laboratory. This climate controlled test facility meets all the demands of Euro 5/6 and SULEV, as well as the stipulations of GTR 15. The vehicles were tested on a chassis dynamometer AVL Zöllner 48". Emissions measurement and analysis was performed using the laboratory's Horiba CVS-CFV sampling system, together with a dilution tunnel Horiba DLT-7020 and a set of Horiba MEXA-7400HRTLE exhaust analyzers for simultaneous measurement of concentrations of carbon monoxide, carbon dioxide, hydrocarbons and oxides of nitrogen. Particle measurement was performed using system HORIBA DLS 7100E and particle counter HORIBA MEXA 2000 SPCS. All the aforementioned devices were controlled by the Horiba VETS-7000NT management system. Sample bags for the sampling of ambient air and exhaust gas were made of Tedlar™ and housed in a heated, insulated unit maintained at 35°C to prevent condensation.

2.3. Procedure

During the tests the tested vehicle were driven over two test cycles: NEDC and WLPC (Fig. 1).

The NEDC test cycle consists of two phases. The first of them simulates urban driving and contains 4 times repeated ECE 15 Urban Driving Cycles (UDC). The second phase – Extra-Urban Driving Cycle (EUDC) reflects highway driving with maximum speed of 120 km/h. The total duration of the cycle is 1180 s, theoretical distance is 10.93

km, and average speed is 33.35 km/h (43.10 km/h excluding stops, which last 267 s). In the NEDC test the speed trace and gearshift timing are fixed and the same for all kind of cars.

The WLTC test cycle is the new speed trace of the new World Harmonized Light Vehicles Test Procedure. According to WLTP, there are 3 WLTC driving cycles: class 1, 2 and 3. The class is chosen taking into account engine power to vehicle mass ratio. All 3 examined vehicles, as practically all European passenger cars, belong to class 3 (max power/kerb weight > 34 W/kg). WLTC class 3 cycle consists of 4 phases: low, medium, high and extra high which simulate successively urban, extra urban and highway driving with max speed of 131.3 km/h. The whole test lasts 1800 s and the vehicle travels a distance of 23.27 km at an average speed of 46.5 km/h (53.5 km/h excluding stops, which last 235 s in total) during this time. The gearshift timing in WLTC is not fixed, but is calculated by a specific computer algorithm for every vehicle taking into account vehicle weight and engine full load power characteristic.

Tests were carried out according to the test procedure applicable to the test cycle in question, but the basic approach was the same. For the vehicles tested in this work, the prescribed gearshifts were used for the NEDC and the calculated gearshifts for the WLTC (different for each vehicle tested). Dynamometer settings has been determined using the alternative methods according to the applicable regulations. However, the vehicle configuration (mass, tire type and pressure, fuel) was the same in NEDC and WLTC tests.

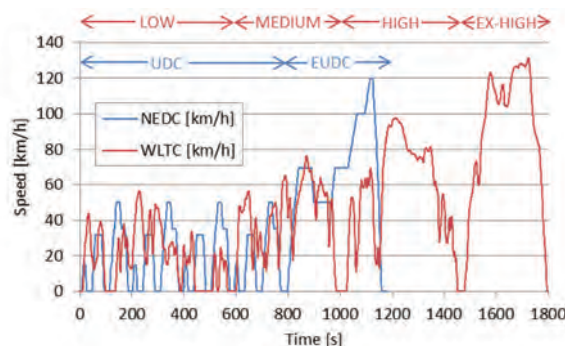


Fig. 1. Speed profile for WLTC and NEDC cycle

3. Results and discussion

3.1. NEDC

Fuel consumption (FC) and emissions of carbon monoxide (CO), unburned hydrocarbons (THC), nitrogen oxides (NO_x) and particle number (PN) obtained in each phase of the NEDC and in the entire cycle for the tested vehicles are presented in Figs 2–6. As expected, the lightest car equipped with the least powerful engine (vehicle C) had the smallest fuel consumption in the NEDC cycle, however the highest fuel consumption had a vehicle B of almost the same mass but equipped with the most powerful engine. Fuel consumption of considerably heavier vehicle A was by 4% lower than fuel consumption of vehicle B, and 9% greater than of vehicle C. For all cars fuel consumption in urban phase (UDC) was much greater than in highway phase (EUDC) – from almost 50% for vehicle A to over

80% for vehicle B. Despite of the heaviest mass vehicle A had only 3% greater fuel consumption than vehicle C and 16% smaller than vehicle B during UDC. On the other hand in spite of the 6-speed gearbox vehicle A had the highest fuel consumption in EUDC (vehicles B and C were equipped with 5-speed gearboxes).

Emissions of toxic compounds during NEDC cycle from particular vehicle tested differ significantly, but it is hard to point at the vehicle with the lowest or highest emission, because it depends on the compound considered (Figs 3–6). Generally, for each of the vehicles, emissions of all toxic compounds are much higher in UDC than during EUDC, what is related to the lower average speed – 18.4 km/h in UDC vs 62.6 km/h in EUDC, and cold engine at the beginning of the UDC. Particularly large differences between results in UDC and EUDC occurred in the case of hydrocarbons, where the THC emission in UDC was at least 50 times higher than in the EUDC for each of the tested cars (Fig. 4). As expected, the emission of particulate matter from the vehicle equipped with a direct injection engine (vehicle A) was higher than that of cars with indirect injection engines (Fig. 6).

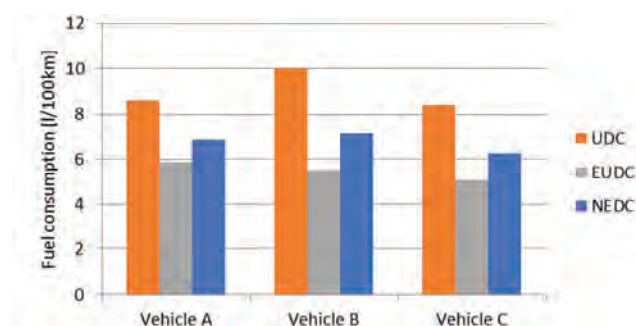


Fig. 2. Fuel consumption in NEDC test and its particular stages

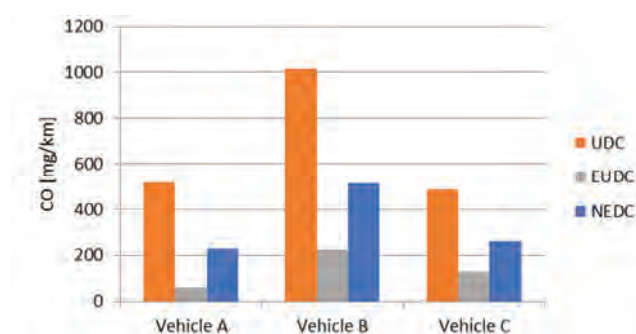


Fig. 3. Carbon monoxide emissions in NEDC test and its particular stages

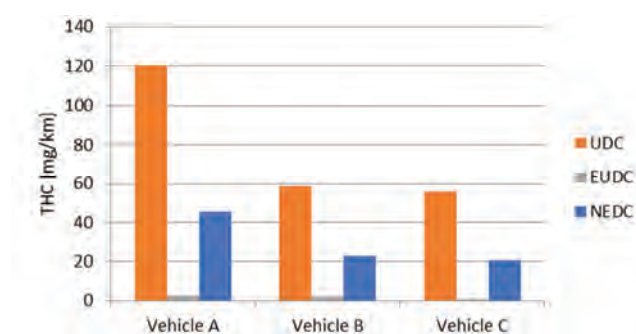


Fig. 4. Hydrocarbons emissions in NEDC test and its particular stages

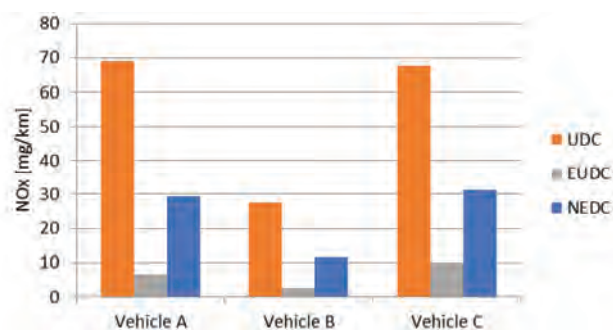


Fig. 5. Nitrogen oxides emissions in NEDC test and its particular stages

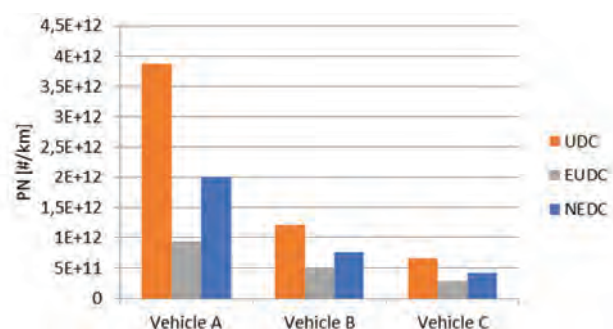


Fig. 6. Particle emissions in NEDC test and its particular stages

3.2. WLTC

Fuel consumption in the WLTC cycle and in its particular stages for all tested vehicle are presented in Fig. 7. The heaviest of the cars (vehicle A) consumed the most fuel, and the lightest one the least (vehicle C). Such trend was also observed for particular phases of the cycle, apart from the low phase, where vehicle B consumed the most fuel. It should be also noted that despite of the 6-speed gearbox vehicle A had much greater fuel consumption than the other cars in extra high phase. All tested vehicles achieved the highest fuel consumption in the first phase of the WLTC, and the lowest in the third.

The emissions of toxic compounds in WLTC cycle and its particular phases are presented in Figs 8–11. As in the NEDC test, it is difficult to identify the vehicle with the highest or lowest emission in the WLPC cycle. Vehicle A had the highest particulate emission, vehicle B the highest CO emissions, and vehicle C the highest NOx emission.

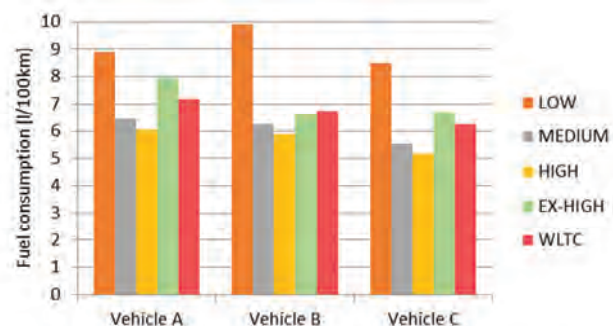


Fig. 7. Fuel consumption in WLTC test and its particular stages

For all cars tested, the emission of all toxic compounds in the first phase of the WLTC was definitely higher than in the remaining ones. The greatest difference, as in the

NEDC, was in the case of hydrocarbons. This was caused by long stop times (27% of the duration of the phase) and low average speed (19 km/h) in this phase, and above all by the heating up of the engines and aftertreatment systems.

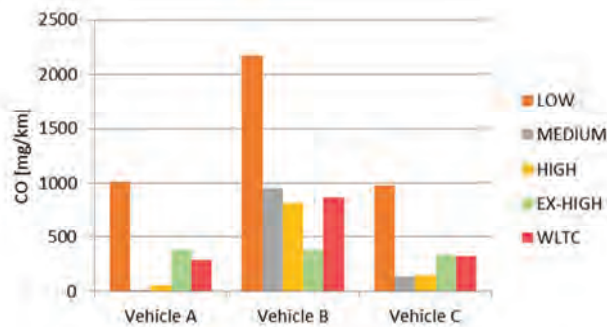


Fig. 8. Carbon monoxide emissions in WLTC test and its particular stages

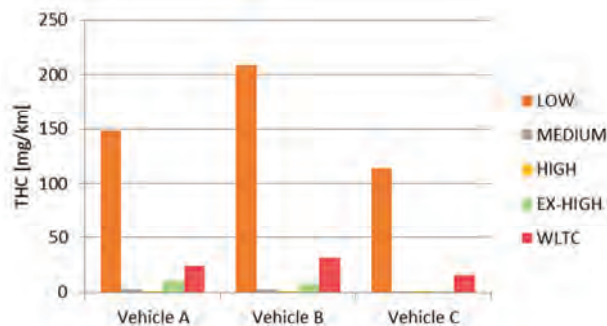


Fig. 9. Hydrocarbons emissions in WLTC test and its particular stages

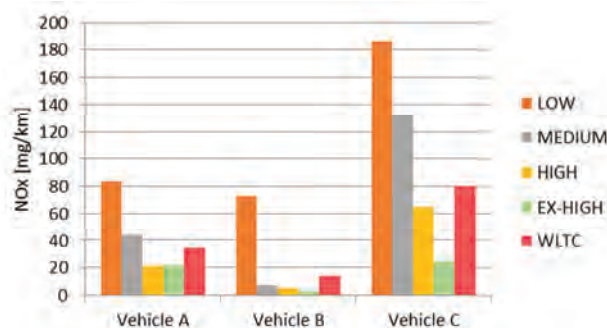


Fig. 10. Nitrogen oxides emissions in WLTC test and its particular stages

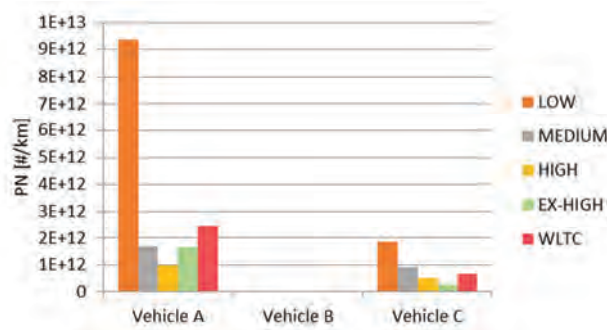


Fig. 11. Particle emissions in WLTC test and its particular stages

3.3. Comparison of results from NEDC and WLTC

The results achieved in the NEDC and WLTC cycles are compared in Figs 12-16. Only vehicle A achieved higher fuel consumption in the WLTC cycle than in the NEDC cycle (by 4.5%). The other two cars had lower fuel consumption in the WLTP cycle – vehicle B by 6% and vehicle C by 1% (Fig. 12).

The emissions of carbon monoxide (CO), nitrogen oxides (NO_x) and particulate number (PN) were higher in the WLTC than in the NEDC, by 27%, 70% and 24%, 17%, 19% and 156%, and 23%, undetermined and 64%, respectively for vehicles A, B and C (Figs 13, 15 and 16). Only vehicle B had lower hydrocarbons (THC) emission in the WLTC, by 53% in comparison to THC emission in the NEDC. THC emissions for the other two cars in the WLTC were lower by 47% and 24%, respectively for vehicle A and C (Fig. 14). The effect of the cycle on particulate matter emission (PM) was determined only for vehicle A and this emission was greater by 10% in the WLTC (Fig. 16). PMs were measured only for entire cycles, not for their particular phases.

It should be noted that all tested cars in both NEDC and WLTC cycles met the requirements of Euro 6 standard for CO, THC and PM emissions – the limits are 1000 mg/km, 100 mg/km and 4,5 mg/km, respectively. In the case of NO_x emission, only vehicle C in the WLTC did not fulfill the requirement – the limit is 60 mg/km (Fig. 15). The Euro 6 PN emission limit of 6×10^{11} #/km has been significantly exceeded by vehicle A and slightly by vehicle B (Fig. 16).

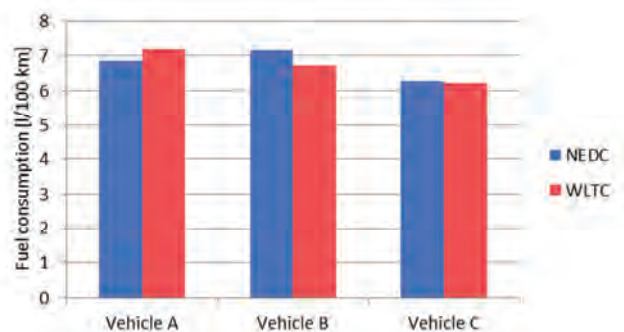


Fig. 12. Fuel consumption in NEDC and WLTC tests

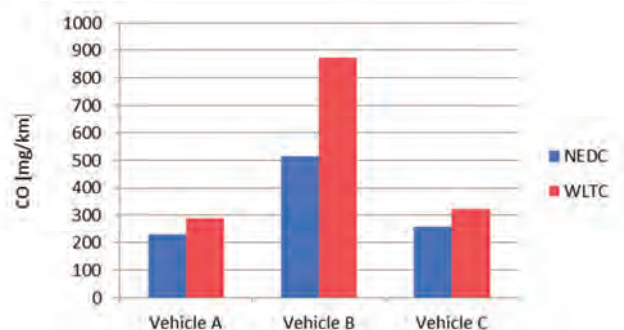


Fig. 13. Carbon monoxide emissions in NEDC and WLTC tests

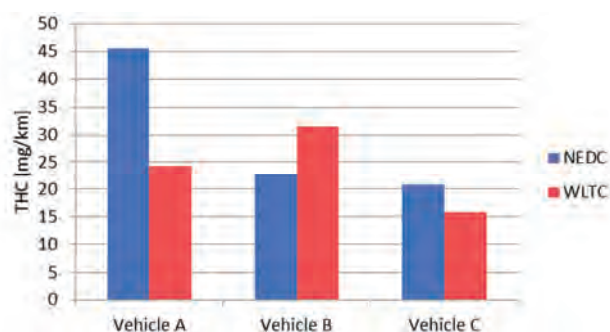


Fig. 14. Hydrocarbons emissions in NEDC and WLTC tests

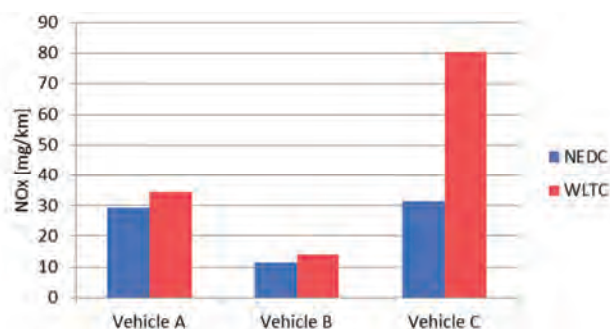


Fig. 15. Nitrogen oxides emissions in NEDC and WLTC tests

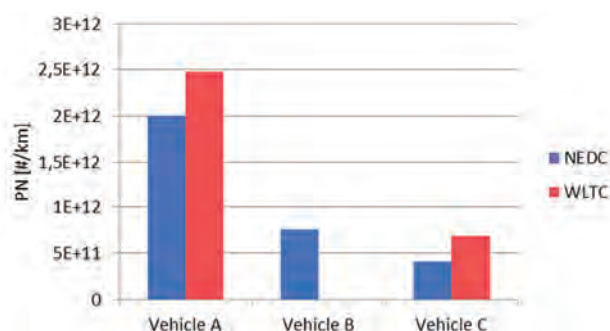


Fig. 16. Particle emissions in NEDC and WLTC tests

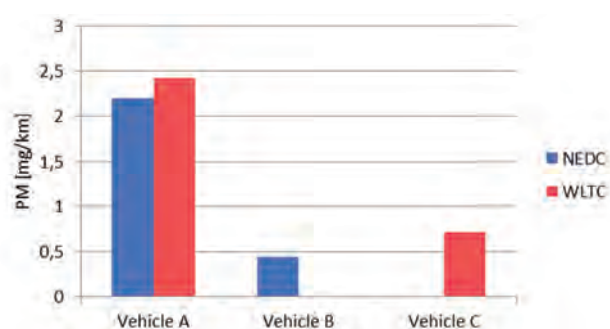


Fig. 17. Particle emissions in NEDC and WLTC tests

4. Conclusions

Measurements of fuel consumption and emissions of regulated exhaust compounds in 3 different passenger cars equipped with gasoline engines of the same displacement 1.4 dm³ but different design solutions (direct and indirect injection, turbocharging and natural aspiration) were carried out. Each of the cars was tested on a chassis dynamometer in the conditions of two driving cycles: the New European Driving Cycle (NEDC) and the Worldwide Harmonized Light Duty Test Cycle (WLTC). In both cycles, a given tested car was equipped and configured in the same way.

The fuel consumption obtained in the new WLTC cycle only in the case of one of the cars tested was 4.5% higher than in the NEDC cycle. In the case of the second car it was lower by 6% and in the case of the third car the fuel consumption were practically the same in both cycles (smaller in WLTC by less than 1%). This indicates that the introduction of a new driving cycle may not significantly increase the fuel consumption obtained in the tests. The introduction of the new Worldwide Harmonized Light Duty Test Procedure (WLTP), which relevant part is WLTC, was intended to bring the results obtained in type-approval tests closer to the results obtained in normal road operation. However, it should be emphasized that the WLTC driving cycle is just one of the factors that distinguishes the new WLTP procedure from the previous NEDC one. Other important factors affecting fuel consumption have not been the subject of this study (WLTP limits the possibilities of unrealistic reduction of vehicle weight, energy consumption by auxiliary equipment, resistance of movement, etc.).

For all cars tested CO, NO_x, PN and PM emissions in the WLTC cycle were higher than in the NEDC cycle, on average for the three examined cars by: 48%, 78%, 30% and 10% respectively.

The average for the three cars' THC emission in the WLTC cycle was lower by 20% compared to the emission in the NEDC cycle. However, in this case the impact of the cycle is ambiguous, because the lower THC emission was obtained for the two cars tested, and for one car the THC emission was higher in the WLTC cycle.

Summarising, it can be concluded that replacing the NEDC with WLTC Class 3 driving cycle itself does not result in higher fuel consumption, and so it does not contribute much to the elimination of the problem of significantly lower type-approval fuel consumption than the actual one. However, the emissions of regulated exhaust compounds determined in the new WLTC driving cycle are higher than emissions obtained in the NEDC cycle, except for THC.

Nomenclature

DI direct injection
 EUDC extra-urban driving cycle
 FC fuel consumption, dm³/100 km
 MPI multi-point injection
 NEDC new European driving cycle
 PM particulate matter

PN particle number
 SI spark ignition
 UDC urban driving cycle
 WLTC worldwide harmonized light duty vehicle test cycle
 WLTP worldwide harmonized light duty vehicle test procedure

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Analysis of the applicability of a common rail pump for an aircraft engine

The paper presents an analysis of the possibility of using a common rail pump to supply an aircraft compression-ignition engine. It is an engine with a two-stroke cycle, three cylinders, opposing pistons and 100 kW power. Its each combustion chamber is supply by one or two injectors controlled by electromagnetic valves. In order to assess the possibility of using a common rail pump, four high-pressure pumps were tested on a bench. They are piston pumps differing in the number and geometry of their pumping sections. The analysis included the pumping output, the torque on the pump drive shaft and the power needed to drive the pump. The weight and overall dimensions of the pump were also considered, including the arrangement of the pumping sections and the way the drive is transmitted. The research allowed to optimize the engine power supply system depending on fuel demand and the way the pump is mounted on the engine.

Key words: diesel engine, two-stroke, opposing piston, fuel pump, common rail

1. Introduction

Demanding requirements to reduce exhaust emissions and fuel consumption are increasingly affecting internal combustion engines for non-motorization applications. At the same time, the aim is to increase the power factor. As a result, intensive research work is underway to develop a diesel-powered engine for the propulsion of aircraft. Due to a number of advantages, such as: lack of the head (less heat loss) and the camshaft system, opposing piston movement conducive to engine balancing, the piston opposed diesel engine operating in a two-stroke cycle was developed and modernized [1–3]. Obviously, such a construction also has drawbacks. The main one is the need to use a gear that connects two crankshafts or a complex crank system with one shaft. The development of these engines so far has been limited by regulations on meeting the emission standards for particulate matter and hydrocarbons [4]. With the development of electronics and materials engineering, these engines are now experiencing a renaissance. The optimistic research results [5–7, 24] intensified the work on the development of efficient two-stroke diesel engines [8, 9, 25]. One of the ways to eliminate unfavorable exhaust emissions while reducing fuel consumption is to use one of the most modern fuel supply systems. The development in this field has allowed the development of a high-pressure, common rail diesel fuel supply system for diesel engines which is widely used in the automotive industry. Compared to previous solutions, it is more flexible in terms of injection, high fuel pressure generation and control and reduced energy demand [10–12].

The literature reports on some research into the injection process and related fuel dynamics [13–15, 23], a maintenance of a given fuel pressure in the tank [16–18] or an injection control strategy [19–21].

An equally important part of the fuel system is the high-pressure pump. In the case of aircraft engines, no new pumps are built, but the proven solutions available on the automotive market are used. The market offer are high pressure pumps, manufactured mainly by Bosch, Denso, Delphi, Siemens. Applying this solution in aviation, the safety aspects of aircraft maintenance and operation need to

be remembered. On the one hand, the smallest and lightest components must be used, but on the other hand, it must ensure safe and reliable operation of the engine. This gives rise to the dilemma of which pump to choose: single-section, small, light and adjusted to the demand, or a large and heavy multi-sectional pump with oversized efficiency. The required pumping output will, of course, depend on the configuration of the fuel system, namely the type and number of injectors.

The paper presents the results of the tests of high pressure pumps for a two-stroke compression-ignition engine with opposing pistons. The engine has three cylinders, power about 100 kW at shaft speed of 3800 rpm. Its each combustion chamber be supply by one or two injectors controlled by solenoid valves. The research allowed us to optimize the fuel supply system in terms of both efficiency and weight as well as the manner it is installed on the engine and power is transmitted.

2. Experimental apparatus and procedure

2.1. Test bench

The research was carried out on a test stand for testing common rail system elements. This is the STPiW3 test stand (Fig. 1) which is equipped with additional control and

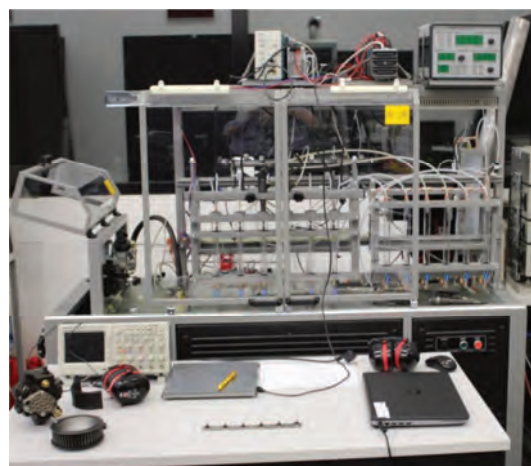


Fig. 1. Common rail system component test bench

measurement systems. Such a test bench enables tests of injectors, high pressure pumps, dosing valves, fuel pressure control valves and pressure sensors. Universal mounting holders allow you to test varied pumps and injectors. High pressure pump is driven with a three-phase 4 kW electric motor controlled by an inverter. The temperature stabilization system enables long-term measurements in constant thermal conditions of the fuel. The diagram of the measuring system is shown in Fig. 2.

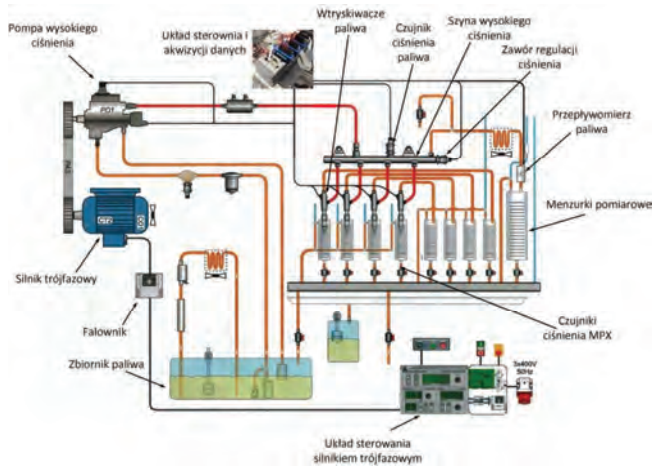


Fig. 2. Diagram of the test bench for testing Common Rail system components [28]

In order to determine the characteristics of high-pressure pumps, the test stand was equipped with control and measurement systems. Control of the common rail system was performed using National Instruments data acquisition cards and the LabView software. This special control and measurement system includes:

- control of actuators:
 - fuel dispenser in the high-pressure pump,
 - fuel pressure regulator in the fuel rail,
- measurements:
 - pump flow rate,
 - fuel pressure in the rail,
 - torque on the pump shaft,
 - pump shaft speed,
- recording and data acquisition system.

To control the actuators, NI-9758 card is used. It allows to control PWM (0–100%) up to four solenoid valves at a maximum current load of 1.5 A (continuous operation), a frequency operation of 2 Hz–10 kHz, a resolution of 500 ns, short-circuit and open circuit detection, supplied from an external source with a voltage of 7–32 V [26].

The control system is based on the CompactRIO-9024 controller. It is a real-time control and data logging device based on a 800 MHz clocked CPU for use in research systems. The platform has a built-in network interface with a data transfer rate of up to 1000 Mbps which enables remote communication and a retrieval of data logged by the device. It has a built-in 4 GB memory and 512 MB DRAM [26].

High pressure pumps are tested with a special program developed in the National Instruments LabView software. Due to hardware requirements, the program had to consist of two modules. The first one was responsible for synchro-

nizing and communicating the actuator cards with the real time controller using DMA channels. The second module, FPGA Xilinx Virtex-5 LX110, is built in a chassis cRIO-9118. It has programmable logic blocks called CLB, I/O type blocks and programmable internal links. The possibilities of programming a particular block and the connections between the blocks are stored in the internal RAM memory.

The voltage measurement signals were recorded with the NI 9205 card. It is a sixteen-channel measuring card with a resolution of 16 bits and a maximum sampling rate of 250 kS/s. The signal level is ± 10 VDC and the absolute accuracy for the range is ± 10 VDC at a full scale of 6.230 μ V [26].

High-pressure pumps were measured with rotor fuel flow meters: DHGF-2 (Fig. 3) and DHGF-4 depending on maximum pump flow rate. Basic technical data of the flow meters is presented in Table 1.



Fig. 3. Flow meter DHGF-2 from Meister [29]

Table 1. Technical data of DHGF series flow meters [29]

Parameter	Unit	Value
Operating pressure, max.	MPa	1.0
Burst pressure (22°C)	MPa	> 3.0
Operating temperature	°C	0–80
Measuring accuracy	%	± 2 of the measured value
Repeatability	%	$\leq \pm 0.8$ of the measured value
Viscosity range	cSt	1–10
Sensing principle		Hall effect, contact-free measuring technique
Process connection	"	Threaded G 1/4 or 5/8 UNF
Power supply	VDC	4.5–24
Output signal		Square wave push-pull output stage
Max. output current (at 24 V)	mA	11
Measuring range		
DHGF-2	dm ³ /h	1.5–100
DHGF-4	dm ³ /h	6–250

To determine the power required to drive the pump, a measuring system was built. Accordingly, the STPiW3 test stand was modernized and equipped with a torque and pump shaft speed measurement system (Fig. 4). The tested pump is mounted on a mounting plate mounted on a frame. The frame and the bearing housings are pin-connected. The axis of rotation of the frame is collinear with the axis of rotation of the pump drive shaft, which enables the transfer of torque from the pump casing to the frame through the plate. This is a universal measuring system capable of testing different pump designs. The pendulum-mounted frame is connected to the station structure by means of a ZEMIC H3-C3-75 kg strain gauge. Torque was measured by measuring the force acting on the spot where the strain gauge is mounted, located 100 mm from the axis of rotation and perpendicularly to the axis of rotation of the pump shaft. Basic technical data of the strain gauge is presented in Table 2.



Fig. 4. Torque measuring system on the high-pressure pump shaft: 1 – high pressure pump, 2 – mounting plate, 3 – frame, 4 – frame pins, 5 – bearing housing, 6 – distance column, 7 – load cells sensor

Table 2. Technical parameters of ZEMIC load cells sensor H3-C3-75 kg [27]

Parameter	Unit	Value
Sensor type		tensometric sensor S-type
Measuring range	kg	≤ 75
Combined error	% FS	$\leq \pm 0.020$
Power supply	VDC	5...12
Output signal	mV	0...20
Terminal resistance, input	Ω	351 $\pm$ 2.0
Temperature range, operating	$^{\circ}\text{C}$	-35...+65

2.2. Scope of the tests

The tests were carried out under the following conditions:

- temperature of the calibration fluid – 40°C ,
- preliminary pump supply pressure – 0.4 MPa,
- high-pressure pump shaft speed – 500–2500 rpm,
- pumping pressure (in the rail) – 0–140 MPa.

The preset pressure was maintained by means of a solenoid valve and was the result of the PID controller. During the tests, each measurement point was defined by pump's shaft rotation speed and the pressure in the rail. The values of pump volumetric flow rate and average torque on the camshaft were recorded four times in 10-second intervals. Additionally, torque, fuel pressure and pump shaft speed with a frequency of 2 kHz were recorded. The average torque value enabled us to specify the power required to drive the pump and then the specific energy demand to obtain the given value of flow rate.

2.3. Subject of the research

The pumps used in the research are widely applied Bosch pumps of different generations and the Denso pump:

1. Bosch CP1_H3,
2. Bosch CP4_1,
3. Bosch CP4_2,
4. Denso HP3.

These are one, two and three section positive displacement pumps with a single or double cam roller. In CP1, the sections are arranged in the body on its circumference every 120° , in CP4.2, they are shifted to each other by 90° and in HP3 by 180° . Compared to the previous solutions (CP1), the CP4 series has an optimized design by reducing the number of components and using an aluminum pump casing. High fuel pressure is generated in the pumping section and then forwarded directly through high pressure lines to the rail. There are no channels in the body for high-pressure fuel flow as it is in the CP1 pump. The same is true for HP3, where the pumping sections are connected by an external steel pipe. The pump body is made of aluminum alloys. Unlike CP pumps, the HP3 pump has a trochoid

type pre-pump. The basic technical data of the tested pumps is presented in Table 3.

Table 3. Technical parameters of high-pressure pumps

Pump name	No. of pumping sections	Max piston stroke	Piston diameter	No. of pumping per rotation	Theoretical flow rate
	[-]	[mm]	[mm]	[-]	[mm ³ /rev]
Bosch CP1_H3	3	5.77	7	3	665.83
Bosch CP4_1	1	7.43	5.5	2	352.87
Bosch CP4_2	2	6.00	6.5	4	795.99
Denso HP3	2	9.00	8.5	2	1020.89
Pump name	drive system	mount	Pre-pump		mass
	[-]	[-]	[-]		[kg]
Bosch CP1_H3	belt transmission	flange	ext. elec.		4.80
Bosch CP4_1	belt transmission	flange	ext. elec.		2.65
Bosch CP4_2	belt transmission	flange	ext. elec.		3.80
Denso HP3	belt transmission	flange	int. trochoid type		3.90

All pumps are equipped with a dosing valve that allows the pump output to be controlled independently of shaft speed. The valve is in the pump casing. The pumps have a flanged mounting and a possibility to place a gear wheel on the pump shaft. This enables us to install the pump in the engine block and drive it by a gear or belt drive.

3. Results and discussion

Table 4 summarizes the required volumetric flow rates of a high-pressure pump. The calculations were made for the engine operation at idle and maximum load conditions and for the option of supplying with three and six injectors (one or two per combustion chamber). The required volumetric flow rate includes injection volume, fuel from leaks and the process of injector control and 25% of the maximum fuel consumption.

Table 4. Required high-pressure pump volumetric flow rates

Type of operation	Injection pressure	No. of injectors		3	6
		Volumetric flow rate			
idle	30 MPa	dm ³ /h		3.45	5.98
max. load	140 MPa			54.37	76.32

Figure 5 presents the analysis of the required volumetric flow rate with the volumetric flow rate of the tested pumps. The horizontal blue lines (idle) and red lines (max. load) illustrate the required flow rate demanded from by a high pressure pump. When the engine is supplied by three injectors, each of the tested pumps has the required volumetric flow rate, but if supplied by six injectors, the CP4.1 pump does not provide such a volumetric flow rate at maximum load.

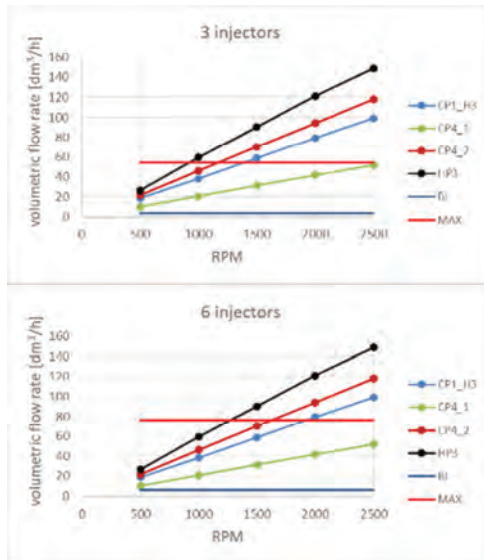


Fig. 5. Comparison of pump volumetric flow rates with the required pump volumetric flow rate – with three and six injectors

The HP3 pump consumes the most power (Fig. 6). At a pumping pressure of 30 MPa, it is from about 0.24 kW (500 rpm) to about 1.5 kW (2500 rpm). However, at a pumping pressure of 140 MPa, it is about 1.0 and 5.0 kW, respectively. The lowest power is consumed by pump CP4.1, and then CP1_H3 and CP4.2. It is respectively about 35 and 76% of the power of pump HP3. The power demand increases linearly as the pump shaft speed increases. Obviously, the differences in the power demand to drive the pump are due to the different geometries of the pumping sections and associated the volumetric flow rates. Therefore, the further part of the paper analyzes the specific flow rates of the pumps.

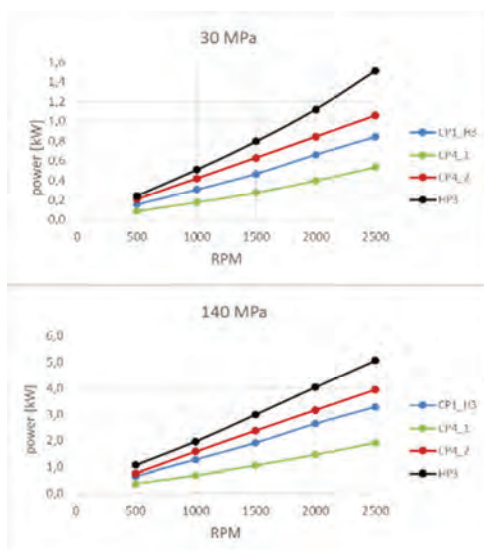


Fig. 6. Comparison of power to pump drive as a function of pump shaft speed for pumping pressures of 30 and 140 MPa

Figure 7 compares the specific pumping rate for pumps operating at a pumping pressure of 30 MPa and 140 MPa. At a pumping pressure of 30 MPa, the largest specific flow rate of approx. 128 dm³/kWh is registered for the CP1.H3

pump, followed by approx. 118 dm³/kWh for the CP4.1 and HP3 pumps, and finally 112 dm³/kWh for the CP4.2 pump. However, the CP4.2 pump keeps the specific flow rate at a similar level throughout the shaft speed range. The other pumps, on the other hand, achieve the maximum specific flow rate in the speed range of 500–1500 rpm, while with increasing speed, the flow rate decreases by about 20%. With pumps operating at maximum pressure, a similar tendency to change the specific flow rate can be observed. However, maximum values of about 30 dm³/kWh for CP1, CP4.1 and HP3 can be observed for 1000–1500 rpm. For higher and lower speeds, the flow rate decreases. For CP4.2, the maximum flow rate is reached at maximum speed. The difference between the max and min values is about 7% for CP1 and CP4.2 and about 15% for CP4.1 and HP3.

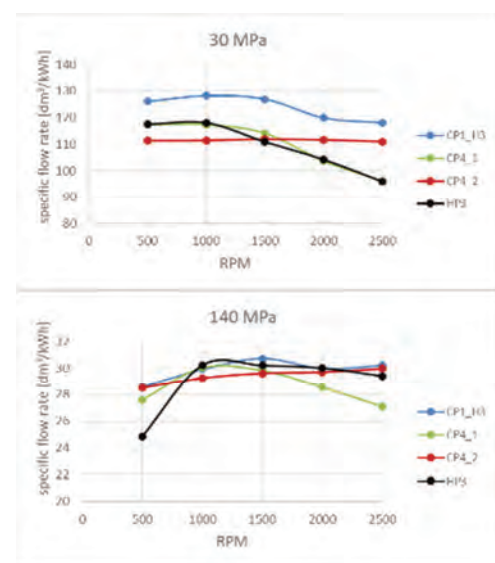


Fig. 7. Comparison of pump specific flow rates as a function of pump shaft speed for pumping pressures of 30 and 140 MPa

Figure 8 shows the torque on the shafts of the tested pumps. This is a sample for a pumping pressure of 30 MPa and 2500 rpm. This figures enabled us to specify the values of the amplitude as well as the minimum and maximum torque on the pump shaft. The calculations were made for the pumping pressures of 30 and 140 MPa.

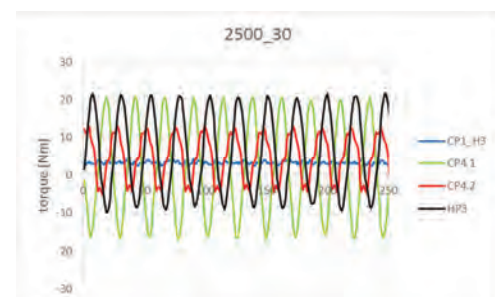


Fig. 8. Torque on the pump shaft at a pumping pressure of 30 MPa and a pump shaft speed of 2500 rpm

The lowest torque amplitude value was registered for the CP1_H3 pump. The torque amplitude analysis (Fig. 7)

shows that the lowest value is obtained for the CP1 pump at both low and high pressures and at low and high speeds (minimum approx. 3 Nm, maximum approx. 28 Nm). A comparable maximum value was also recorded for CP4.2. The highest torque amplitude values can be observed for CP4.1 and HP3. For a pumping pressure of 30 MPa, it is about 39 Nm, and for a pumping pressure of 140 MPa about 59 Nm.

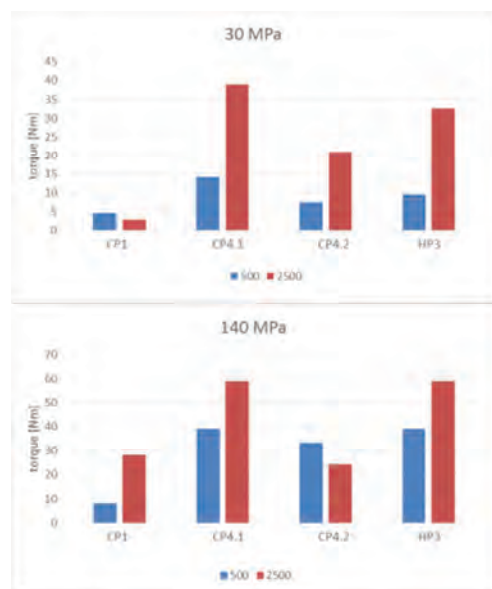


Fig. 9. Torque amplitude on the pump shaft at a pumping pressure of 30 and 140 MPa and pump shaft speeds of 500 and 2500 rpm

Figure 10 and Fig. 11 show the minimum and maximum torque values in relation to the pumping pressure and the pump shaft speed. The most regular torque, mostly of positive or slightly negative (approx. -3 and -7 Nm) values, over the entire operating range is for the CP1 and CP4.2 pumps. The second group is the CP4.1 and HP3 pumps. The torque takes both positive and negative values. The CP4.1 pump has higher negative values of approx. -25 Nm at high speeds. The highest positive torque of the HP3 pump (about 53 Nm) at 140 MPa and 2500 rpm.

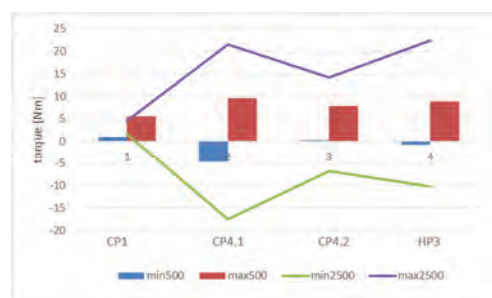


Fig. 10. Minimum and maximum torque at 30 MPa pumping pressure and 500 and 2500 rpm shaft speeds

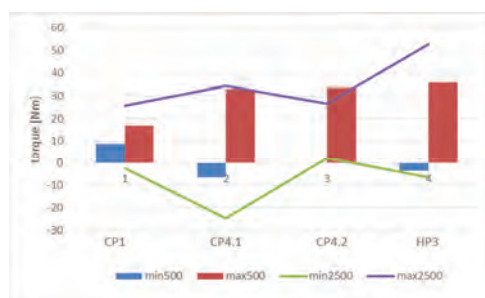


Fig. 11. Minimum and maximum torque at 140 MPa pumping pressure and 500 and 2500 rpm shaft speeds

4. Summary

The tests show that to power an engine of a power of about 100 kW and with three injectors (one per combustion chamber), all three pumps have the required flow rate, both at idle and maximum load conditions. If six injectors (two per combustion chamber) are used, the CP4.1 pump does not provide the required capacity at maximum load.

Analyzing the specific flow rate, it can be stated that there are smaller differences between the tested pumps at the maximum pumping pressure. It is also observed that the CP4.2 pump has an almost constant the specific flow rate in the entire speed range. The efficiencies of the other pumps are highest in the range of 500–1500 rpm.

The smallest torque amplitude is recorded for the CP1 pump. There is a positive torque almost over its entire operating range, which ensures that the drive system is loaded with one-way forces. Another advantage is the number of pumping sections, allowing the synchronization of pumping and fuel injection. However, its large mass and no individual high-pressure connections disqualifies it from mounting in the aircraft engine.

The best option is the CP4.1 pump due to its small mass. However, it has one pumping section, which cannot guarantee the required safe operation of aircraft. With this in mind, a pump with a minimum of two pumping sections and pressure individually discharged into the fuel rail/rails can be applied, which makes the system more reliable. Pumps CP4.2 and HP3 satisfy these requirements. The required flow rate is achieved in the speed range of 1500–2000 rpm. This allows the use of a 1:2 pump drive ratio in relation to the engine crankshaft, reduction of rotation and operation with the highest efficiency (specific flow rate). The pumps are flanged and can be fitted with a toothed wheel or a belt. Accordingly, a toothed or belt transmission can be used and the pump can be more flexibly mounted on the engine with other engine equipment. The advantage of HP3 is the opposite position of its pumping sections and a built-in pre-feed pump. The disadvantage of the pump is that the flow rate decrease with increasing speed (especially at 30 MPa) and higher torque amplitude.

Acknowledgements

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Comparison of exhaust emission from the most commonly used aircrafts with implementation LTO cycle to operating conditions

The LTO cycle (Landing and Take-off cycle) is a research method used for jet engines certification. The measurements are performed in stationary conditions. With the development of aviation, the LTO test started to be carried out more often, but not for its' original purpose. A new aim was an assessment impact of aircraft movement on environment in airports area. LTO cycle consists of four phases (according to ICAO Annex 16): take-off, climb out, approach and taxi/idle. Every airport has different infrastructure, what affects on time during basic flight operations especially for a taxi phase. To rate exhaust emission from aircrafts during basic fly operations duration time should be adjusted to every single airport. As a research area Poznań-Ławica Airport which is located in Poland, was adopted. Based on parameters calculated specially for this airport it is possible to computed there emission from the mostly used aircraft.

Key words: LTO cycle, toxic exhaust gas emission, airlines fleet, adaptation, operating conditions, LTO adaptation

1. Introduction

Air transport is the one of the most rapidly developing branches of transport. The increase in the demand for air transport is caused by the competitiveness of this branch primarily in terms of transport time and safety [1]. The Airbus Global Market Forecast indicates that the air traffic doubles every 15 years [2]. It means that as time passes, more and more aircraft will be flying. The issue of the environmental impact of aircrafts is included in Annex 16 of the Chicago Convention, which applies to countries belonging to the International Civil Aviation Organization (ICAO) [3]. It proposed a universal procedure used to assess the emission of harmful exhaust gases from civil aircraft engines. The LTO (Landing and Take-off cycle) is a research method used for jet engines certification. The measurements are performed in stationary conditions. The individual phases of the test correspond to the generalized time of approach, taxiing, take-off and climb out of the aircraft. For civilian airplanes there are four basic test phases (Fig. 1): start – 100% Fc max (maximum thrust), climb – 85% Fc max, landing – 30% Fc max, taxiing – 7% Fc max [4]. The entire standard LTO test takes about 30 minutes.

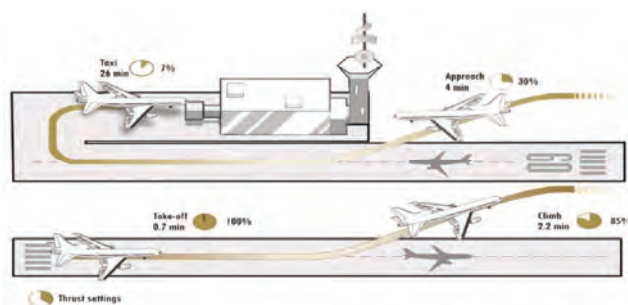


Fig. 1. Landing and Take off cycle [7]

With the development of aviation, the LTO test started to be carried out more often, but not for its' original purpose. A new aim was an assessment impact of aircraft movement on environment in airports area. Each of LTO phases them assigns different thrust and duration time. This methodology is unmatched to conditions prevailing on airports.

Analyzing regulations it should be noted, that duration time do not reflect real conditions during each phase. Every airport has different infrastructure, what affects on time during basic flight operations, especially for taxi phase. Duration time should be adjusted to every single airport. By adjusting the LTO cycle to a given airport, It can be calculated the emission of harmful compounds. In the article, based on the calculated lengths of individual LTO phases tailored to the selected airport, emissions from aircraft most commonly used on the Polish market were calculated.

2. Research methodology

As research area Poznań-Ławica Airport was adopted, which is located in Poland. Using the times of the individual phases of the LTO test calculated for the airport in Poznan (Table 1), it was possible to calculate the emission from the most commonly used aircraft. The times of individual LTO phases, for this airport, was calculated in research carried out in article [5], which is focusing on implementation LTO test in flight conditions. The research were carried out using FNPT II MCC simulator.

Table 1. List of regulations and adopted parameters in the LTO cycle [5]

Phase	Thrust	Duration time [min]	
		regulations	adopted
Take off	100%	0.7	0.67
Climb out	85%	2.2	1.88
Approach	30%	4	5
Taxi/Idle	7%	26	9.9

Aircrafts, which are mostly used at Poznań-Ławica Airport are: Boeing B737-400, B737-800 and Airbus A320 and A321. These models were chosen based on fleet of airlines with the largest traffic at Polish market, belong to LCC (*Low cost carriers*). First aircraft is Boeing 737-400 – the classic model. It is the world's most popular medium-sized passenger airliner. It is produced in the United States. Models 737-400 were produced in the years 1983-2000 and it takes on board from 146 (8 in the premium class and 138 in economy) to 188 passengers. Its maximum cruising speed is 912 km/h [8].

Table 2. Described of aircrafts chosen to the analysis [8–11, 13–16]

Type of Aircraft	Wing-span [m]	Maximum speed [km/h]	Number of passengers takes on board
Boeing 737-400	28.88	912	146–188
Boeing 737-800	28.9	840	162–189
Airbus A320	34.1	871	up to 180
Airbus A321	34.1	871	up to 220

Next aircraft model is Boeing B737-800. In the basic version, the model 737-800 takes between 162 and 189 passengers on its board. It reaches of 12,496 m, a speed of up to 840 km/h and a no-landing range of 5420 km. At the end of February 2016, in Shanghai, Boeing presented the new machine model 737-800, which is the transport version marked as Boeing 737-800 BCF (*Boeing Converted Freighter*) [12].

Airbus A320 is the most popular model from the entire A320 family. It is the most frequently chosen plane by low-cost airlines, however, its largest operators are traditional, large and medium-sized carriers. This aircraft was produced in both 100 and 200 versions, and the differences were mainly in the length of the range. The first users were Air France and British Airways. The A320 is the first designed aircraft from the entire A318 / A319 / A320 / A321 family [10].

The last aircraft, which is included in analysis is a medium-range passenger plane manufactured by the European Airbus consortium in Toulouse. The Airbus A321 is the largest of the Airbus A320 Family series and the world's largest medium-range passenger plane carrying over 200 passengers. Airbus A321 is a competitor for the American Boeing 737-900 [11].

According to times of the individual phases of the LTO cycle calculated for the Poznań-Ławica Airport it should be noted, that calculated duration time each of LTO phases (Fig. 1) is most different at two phases: approach and taxi/idle. The approach phase is 25% longer than in regulations and taxi/idle phase is 62% shorter than in regulations [5]. It can significantly affect to emissions at airport area. Based on parameters calculated specially for this Airport (Table 1) it is possible to computed emission from the mostly used aircraft.

Methodology was concern on computed toxic exhaust fumes emission using the LTO cycle. Emission in LTO cycle is calculated using formula [1]:

$$EPC_{pol,mode} = (TIM \cdot 60) \cdot FFR \cdot EF \cdot NE$$

where: $EPC_{pol,mode}$ – emissions per cycle for a particular mode [g/phase], TIM – Time in Mode [s/phase], FFR – Fuel

Flow Rate [kg/s], EF – Emission Factor [g/kg], NE – Number of engines on aircraft [–].

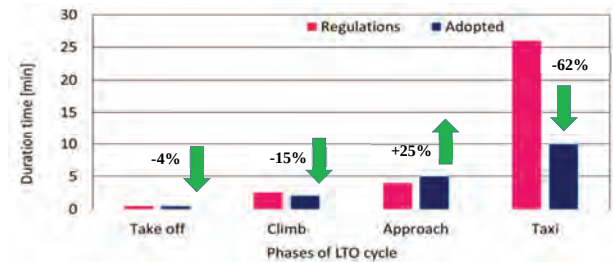


Fig. 6. Differences between LTO time phases in regulations and adopted [based on 5]

Table 3. Emission factors for chosen type of aircrafts [6]

Type of aircraft	Emission indicators g/kg				FF kg/s
	Phase	HC	CO	NO _x	
B737-400	Take off	0.036	0.9	19.4	1.056
	Climb out	0.047	0.9	16.7	0.878
	Approach	0.073	3.4	8.7	0.314
	Idle	1.75	30.1	4.1	0.119
B737-800	Take off	0.02	0.25	21.79	1.213
	Climb out	0.02	0.16	17.08	0.986
	Approach	0.05	3.07	8.93	0.331
	Idle	1.75	30.94	4.27	0.108
A320	Take off	0.23	0.9	24.6	1.051
	Climb out	0.23	0.9	19.6	0.862
	Approach	0.4	2.5	8	0.291
	Idle	1.4	17.6	4	0.1011
A321	Take off	0.1	0.9	33	1.051
	Climb out	0.2	0.9	26.2	0.862
	Approach	0.5	2	10.7	0.291
	Idle	3.7	19.8	4.5	0.1011

Based on Aircraft Engine Emissions Databank from 2018 year (Table 3) calculated emission toxic exhaust fumes for following aircrafts: B737-400, B737-800, A320 and A321. Calculations were carried out for emissions in regulations LTO cycle and adopted LTO cycle. The results of calculated emission presented in tables from 4 to 7.

Table 4. Emission toxic exhaust fumes for Boeing 737-400 in g/phase

Emission	Adopted LTO			Regulation lto		
	HC	CO	NO _x	HC	CO	NO _x
Take off	3.1	76.4	1647.1	3.2	79.8	1720.9
Climb out	9.3	178.3	3307.9	10.9	208.6	3870.9
Approach	13.8	640.6	1639.1	11.0	512.4	1311.3
Idle	247.4	4255.3	579.6	649.7	11175.5	1522.2
Sum	273.5	5150.5	7173.7	674.8	11976.4	8425.3

Table 4 contains the results of calculations for emissions, toxic exhaust components, performed in g/phase. Analyzing the results contained in the table, it can be seen that the B737-400 aircraft emits more carbon oxides in regulation LTO test. According to the adopted calculation methodology, it turns out that in the case of adopting the LTO test model based on times adapted for the selected airport, the CO emission is then about 2 times less. A simi-

lar situation applies to HC. In the case of nitrogen oxides, the emission in both tests is comparable. Regarding the emissions per test phase, the CO emissions increase in the idle phase, and NO_x in the climb out phase.

The largest NO_x emission in this phase results from the engine's operating conditions at the moment of climbing.

Table 5. Emission toxic exhaust fumes for Boeing 737-800 in g/phase

Emission	Adopted LTO			Regulation LTO		
	HC	CO	NO_x	HC	CO	NO_x
Take off	2.0	24.4	2125.1	2.0	25.5	2220.2
Climb out	4.4	35.6	3799.3	5.2	41.6	4446.0
Approach	9.9	609.7	1773.5	7.9	487.8	1418.8
Idle	224.5	3969.7	547.9	589.7	10425.5	1438.8
Sum	240.9	4639.4	8245.7	604.9	10980.4	9523.8

Analyzing the data in Table 5, it can be seen that the Boeing 737-800 produces less toxic exhaust fumes compounds than B737-400. As in the previous case, the largest emission concerns on carbon oxides. In the case of the adopted LTO, the CO emission, is much smaller than in regulations test. On the other hand, hydrocarbons are almost three times less. In the case of nitrogen oxides, the emission in both tests is comparable. Regarding the emissions per test phase, the largest emission occurs in phases as in the previously discussed case.

Table 6. Emission toxic exhaust fumes for A320 in g/phase

Emission	Adopted LTO			Regulation LTO		
	HC	CO	NO_x	HC	CO	NO_x
Take off	19.4	76.1	2078.7	20.3	79.5	2171.8
Climb out	44.7	175.0	3811.6	52.3	204.8	4460.3
Approach	69.8	436.5	1396.8	55.9	349.2	1117.4
Idle	168.1	2113.9	480.4	441.6	5551.6	1261.7
Sum	302.2	2801.5	7767.5	570.1	6185.1	9011.3

Based on the data in Table number 6, it can be seen that the largest emission concerns nitrogen oxides. As in previous cases, the CO emission is almost 3 times higher in the case of a standard LTO test. On the other hand, hydrocarbons are almost half less when using the LTO test developed for the selected airport. Regarding the emissions per test phase, the largest emission for Airbus A320 occurs in the same phases as in the previously discussed case.

Table 7. Emission toxic exhaust fumes for A321 in g/phase

Emission	Adopted LTO			Regulation LTO		
	HC	CO	NO_x	HC	CO	NO_x
Take off	8.5	76.1	2788.5	8.8	79.5	2913.4
Climb out	38.9	175.0	5095.0	45.5	204.8	5962.3
Approach	87.3	349.2	1868.2	69.8	279.4	1494.6
Idle	444.4	2378.1	540.5	1167.1	6245.6	1419.4
Sum	579.0	2978.4	10292.3	1291.3	6809.2	11789.7

Table 7 contains emission values of toxic components of the Airbus A321 fumes. As in the previous model, this plane emits the most nitrogen oxides. This model produces almost twice as many hydrocarbons than the Airbus A320.

3. Analysis and research result

3.1. Analysis for Boeing B73-400 and B737-800

The results of the calculations carried out are presented in the form of diagrams to show the individual dependencies. The sub-section presents and discusses the results for the B737-400 and B737-800. The analysis can be related to approximate course of changes in emission factors: EICO, EIHC, EINO_x , as a function of the Thrust (Fig. 7).

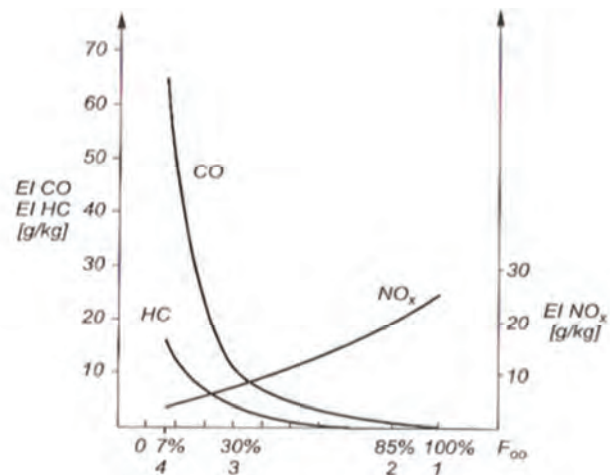


Fig. 7. Approximate course of changes in emission factors: EICO, EIHC, EINO_x , as a function of the Thrust [5]

It can be seen, that emission factor for CO is much bigger in taxi and approach phase, than in take-off and climb out phase. The reason of this, is incomplete combustion and local oxygen deficiency. This conditions occur in taxi and approach phase. Which means that CO emission should be bigger in taxi and approach phase in both cases – adopted LTO and regulations LTO. The same situations consider HC emission but hydrocarbons emission factor is much smaller than CO factor. Conditions, which cause big HC emission are incomplete fuel burn. The opposite situations are in NO_x cases. NO_x emission should be bigger in Take-off and climb out phase and small in taxi and approach phase. This situation is caused by high temperature and pressure in aircraft engine, which occur in take-off and climb out phase, because it's necessary to use the maximum – 100 % and almost maximum – 85% Thrust in this phases. The analysis should confirm these stated and also demonstrate the differences in results, received in calculation carried out in regulations and adopted LTO cycle.

Figure 8 shows the emission of hydrocarbons in g/phase in individual phases of the LTO test with the comparison of the emission volumes obtained in LTO regulations and in the LTO adopted for B737-400 and B737-800 for Poznań-Ławica Airport. It can be seen, that there is a significant difference in the idle phase for LTO adopted results, HC emission is almost 3 times lower than in the case of standard LTO. It can also be seen that the largest emission of hydrocarbons falls on the Idle phase, which is caused by the longest duration of this phase and the conditions prevailing in the engine. The emission of carbon oxides is much higher than in the case of HC emissions. The biggest difference in emissions can be seen in the Idle phase.

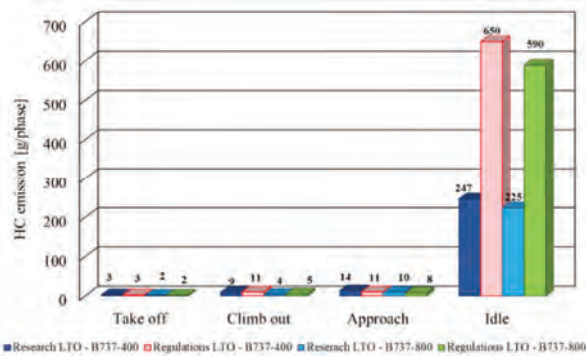


Fig. 8. HC emission for B737-400 and B737-800 in regulations and research LTO cycle in each phase

In the case of regulations LTO, this emission is almost 3 times higher than in the case of LTO referees. A large emission of carbon monoxide also occurs in the Approach phase. In all phase of the test the CO emission is higher for the B737-400 model. Another chart created is CO emissions in g/phase, as previously for the two Boeing aircraft models, presented for each phase of both versions of the LTO cycle (Fig. 9).

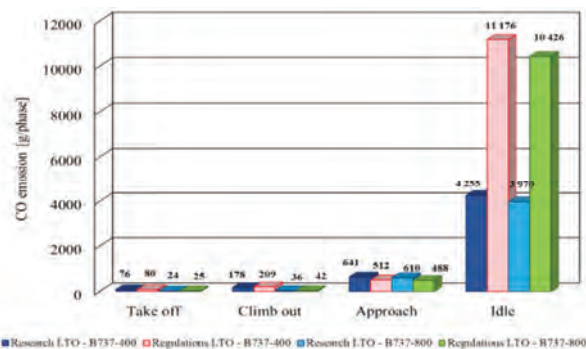


Fig. 9. CO emission for B737-400 and B737-800 in regulations and adopted LTO cycle in each phase

Figure 10 performed the NO_x emission for the aircraft models in question. The emission of nitrogen oxides differs significantly from other toxic exhaust components. The largest emission is in the climb out phase, because in this phase almost entire engine power is used, and NO_x are formed at very high temperature conditions.

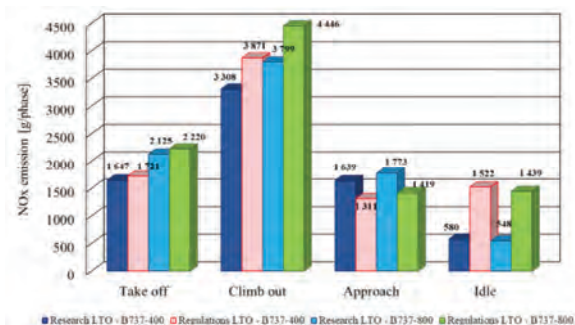


Fig. 10. NO_x emission for B737-400 and B737-800 in regulations and adopted LTO cycle in each phase

In the case of emissions of nitrogen oxides, the results are not as diverse as in the case of hydrocarbons and carbon

monoxide. Diversification concerns not only the value of emissions between particular phases, but also the emission of individual fumes exhaust gases in a given phase. The high NO_x emission value can also be seen in the take-off phase, it is also caused by the high-power demand during the start operation. Figure 11 performed the total emission of individual toxic exhaust components in a standard LTO test and in the adopted LTO. It can be seen, that in the case of hydrocarbons, the difference in total emissions is as high as 60%, while for carbon monoxide, 57%. The smallest difference between the total emissions in calculated LTO tests is for NO_x and it is 15%. The difference is mainly due to the changes in times accepted in the adopted LTO, which was prepared especially for Poznań-Ławica Airport. It can also be said that the largest CO emission occurs during the entire test.

Another issue discussed here is the total emission of toxic exhaust components in both LTO tests for the B737-800 aircraft (Fig. 12).

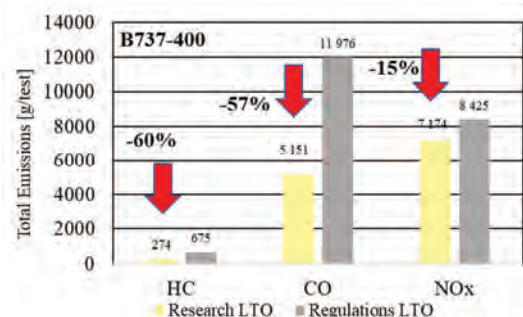


Fig. 11. Total emission of toxic exhaust fumes for adopted and regulations LTO for B737-400

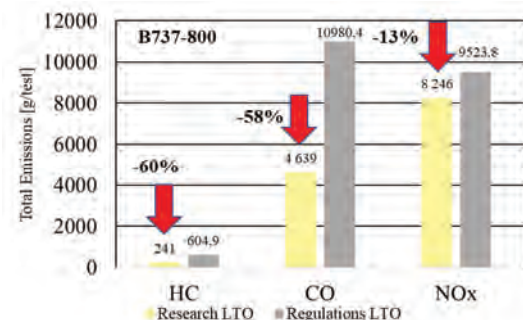


Fig. 12. Total emission of toxic exhaust fumes for adopted and regulations LTO for B737-800

The differences between the two tests are similar to those in the previous case. For the B737-800 aircraft, the total hydrocarbon emission in the LTO test was 60% lower, while in the case of carbon monoxide it was 58%. The smallest difference occurs with the total emission of nitrogen oxides and amounts to 15%. Emissions of individual toxic exhaust components are in the case of B737-400 and B737-800. In the case of the younger model – B737-800, the producers managed to reduce HC and CO emissions to a small extent, while the NO_x emission increased, which most probably results from the highest power of engines fitted on the B737-400 model, and what is associated with it, the higher temperatures prevailing in the engine.

3.2. Analysis for Airbus A320 and A321

The results of the calculations are presented below, in the form of graphs for Airbus A320 and A321 aircraft (Fig. 12). In the case of a hydrocarbon emission characteristic, the highest HC emission occurs in the Idle phase of the LTO test. Analyzing the graph, it can be seen that the higher emission is in the A320 model. In addition, the emission of hydrocarbons in the idle phase for A320 is more than 5000 g/phase higher with the standard LTO cycle than in the case of the adopted LTO. Analyzing the HC emission in the remaining phases of the test, it should be noted that the emission from the A320 model is in any case higher. The smallest hydrocarbon emission occurs in the Take-off phase of the LTO cycle. The emission volume, in each phase, between individual LTO tests is significant for the Airbus A320 model, while in the case of A321 (except the Idle phase) the difference in results are insignificant.

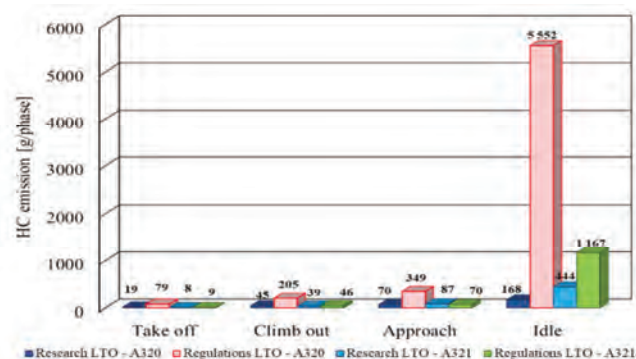


Fig. 12. HC emission for Airbus A320 and A321 in regulations and adopted LTO cycle in each phase

Figure 13 shows carbon monoxide emissions in each phase of both LTO test for Airbus A320 and A321. As in the previous case, the highest emission occurs in the Idle phase, however, it concerns the model of A321, not A320, where the emission of carbon monoxide is the highest in the Climb out phase. It should also be noted that the emissions in the climb out phase, in the case of the A320 model, is almost 30 times higher in the calculations carried out in accordance with the standard LTO test than in the adopted. In addition, there is a large difference in the CO emissions for the A320 model between the standard LTO cycle and the adopted LTO cycle. The next discussed characteristic (Fig. 14) is the emission of nitrogen oxides in the individual phases of the standard and test LTO test for the Airbus A320 and A321 models. The results presented in the chart are similar to those obtained in the Boeing B737-400 and B737-800 analysis. The largest amount of NO_x is produced by Airbus A321 in the Climb out phase.

This is the phase during which the most nitrogen oxides are generated. The volume of emissions also reaches high values in the remaining test phase. While the HC emission was small, apart from the Idle phases, the NO_x characteristic makes it clear that in each phase the emission is significant. The main reason for such a large emission of nitrogen oxides in the Take-off and Climb out phases is the high temperature in the engine during these phases. The high

temperature is caused by the amount of power that must be used during these two phases.

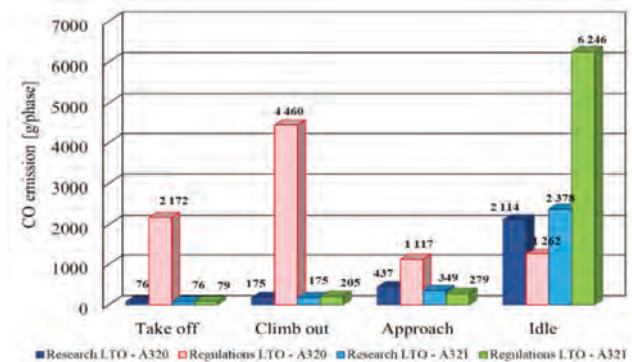


Fig. 13. CO emission for Airbus A320 and A321 in regulations and adopted LTO cycle in each phase

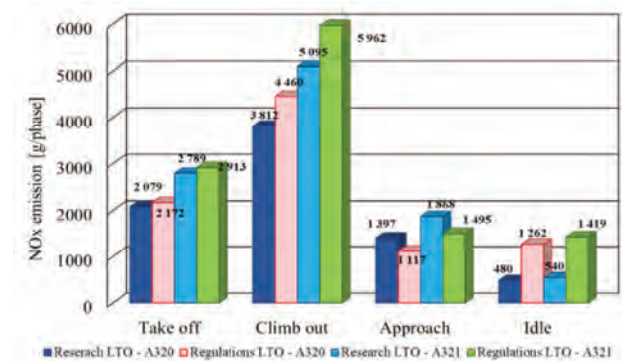


Fig. 14. NO_x emission for Airbus A320 and A321 in regulations and adopted LTO cycle in each phase

At Figure 15 presented the total emission of toxic exhaust components in the standard and the LTO test for the Airbus A320. It can be seen, that in the case of hydrocarbons, the difference in total emissions is not as high as for Boeing, because reach about 47%. For carbon monoxide the differences are about 55%, which means almost the same as in Boeing emissions. The smallest difference between the total emission in calculated LTO tests is for NO_x and it is 14%. The difference is mainly due to the changes in times adopted in the LTO test, which was prepare especially for Poznań-Ławica Airport. It can also be said that the largest NO_x emission occurs during the entire test. Another issue discussed is the total emission of toxic exhaust components in both LTO tests for the A321 aircraft (Fig. 16). The differences between the two tests are similar to those in the previous case.

For the A321 aircraft, the total hydrocarbon emission in the LTO test was 54% lower, while in the case of carbon monoxide it was 56%. These results are similar to results for Boeing analysis. The smallest difference occurs with the total emission of nitrogen oxides and amounts to 13%.

It can be seen, that in the case of the old model – Airbus A320 emission of exhaust gas fumes are smaller than for model A321. It can be cause for the size of models. Airbus A320 is larger and heavier than A321 which can entail bigger emission of exhaust gas fumes.

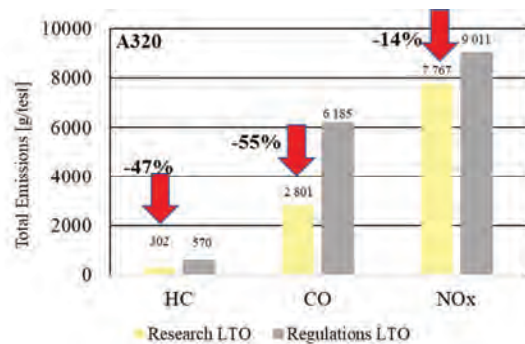


Fig. 15. Total emission of toxic exhaust fumes for adopted and regulations LTO for A320

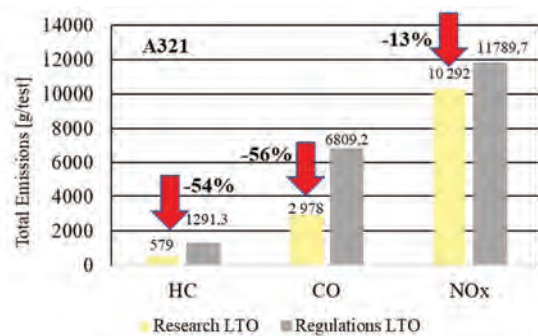


Fig. 16. Total emission of toxic exhaust fumes for adopted and regulations LTO for A321

4. Conclusions

Analysing the characteristics, it should be noted that the change in the time of individual phases of the LTO cycle has a significant impact on the volume of emissions from particular types of aircraft. For HC and CO, emissions are reduced

by 60%. Analyzing the NO_x emission, the differences are smaller, which is caused by the highest emission of this component during take-off, and in this phase the duration time was slightly reduced. The adopted confirms the validity of the thesis that parameters of flight phases adopted in the LTO test performed for the purpose of testing emissions in airport areas are not equal. For this type of adopted, individual parameters should be special for each airport to get results much closer to the actual local emission. The results confirm the suitability of adapting the LTO test to the airport conditions, in order to more thoroughly research and analyze the emissions of toxic gas compounds in the area of airports. Despite the choice of different aircraft models, the results are close to each other. Boeing aircrafts emit more CO than Airbus aircrafts, while the latter emit much more NO_x. On this basis, it is impossible to clearly determine which of the models is less conducive to polluting the environment, because each of them has its advantages and disadvantages. Significant differences occur in the emissions of adopted LTO. The presented methods performed that the emission in the case of Poznań-Ławica Airport is actually smaller than that obtained in the standard LTO test. It should be made aware of the appropriateness of adapting the LTO test to the real conditions prevailing at airports, in order to obtain more real results. There are significant differences in the case of the Poznań-Ławica airport, which is a medium-sized airport. It should be consider what impact the LTO cycle adjustment will have on results in the case of airports like London – Heathrow, Paris – CDG, Amsterdam – Schiphol, Frankfurt Rhein-Main Airport or Barcelona-El Prat. This area is a field for future considerations.

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The influence of cetane-detergent additives into diesel fuel increased to 10% (V/V) of RME content on energy parameters and exhaust gas composition of a diesel engine

To avoid the negative effects of increasing the amount of RME in the diesel fuel (to 10%), three different additive packages were used: stabilising, cleaning, and increasing the cetane number with different concentrations. The tests were carried out using a 4-cylinder, turbocharged 1.9 TDI engine from VW. The tests were carried out for 4 fuels (comparative fuel with a content of 7% RME and 3 test fuels with a content of 10% RME, differing in the content of the additive package). It was found that each of the 3 additive packages used does not have a significant impact on fuel consumption. However, a different effect of the tested additives on the composition of exhaust gases was observed. The first package had a slight effect on reducing the NO_x concentration in the exhaust, but only for small engine loads. On the other hand, the second additive pack worked more effectively only at higher engine loads (in relation to the reduction of NO_x concentration in the exhaust gases). In the third packet, the amount of the cetane additive was doubled (compared to the second packet). Then, the reduction in the NO_x concentration in the exhaust gas by 3–8% was obtained with reference to the comparative fuel.

Key words: cetane-detergent additives, RME, exhaust gas composition, diesel engine

1. Introduction

According to EU preferences, for well-known reasons, the aim is to increase the rapeseed methyl ester (RME) content in the diesel fuel standard. Currently (2019) the mandatory RME add-on to diesel fuel, which is sold at petrol stations in Europe, is 7%. The fuel used in the tests contained 10% of RME. Unfortunately, however, an increase in the amount of methyl esters of vegetable oils is related to i.e. a decrease in the stability of such fuel during storage, increased pollution of the combustion chamber and fuel injector, and increased NO_x concentration in the engine exhaust. Then the increase in NO_x emissions in the exhaust gases is mainly the result of the influence of RME on the elongation of the self-ignition delay and what is connected with the increase in the combustion dynamics and consequently the maximum combustion temperature. Therefore, to avoid the mentioned negative effects of increasing the amount of RME in the fuel (to 10%), three different additive packages were used: stabilising, cleaning, and increasing the cetane number with different concentrations. The tests were carried out using a 4-cylinder, turbocharged 1.9 TDI engine from VW. The tests were carried out for 4 fuels (comparative fuel with a content of 7% RME and 3 test fuels with a content of 10% RME, differing in the content of the additive package). For all the fuels, the measurements were carried out at constant engine speed and 6 engine loads.

2. Characteristics of used and tested fuels

Three Innocet[®] detergent-dispersing packages (A, B, and C) were used. The B10 diesel fuel (10% (by vol.) of RME), whose physico-chemical characteristics are shown in Table 1, was enhanced with the A, B, and C Innocet[®] packages in the quantity of 1,500 mg/kg. Such enhanced B10 diesel fuel had cetane number equal to 58. The Innocet[®] A, B, and C packages contained 50 mg/kg of a commercial detergent-dispersing additive and differed by

the level of dosing of the stabilizers taking synergistic part in creation of free radicals in the area of low-temperature and indirect oxidation of the B10 diesel fuel. In this manner, 4 fuels were created, which then underwent engine tests:

- TM/134/18 – comparative fuel B7 (7% (V/V) RME)
- TM/135/18 – B10 fuel with the Innocet[®] A package
- TM/136/18 – B10 fuel with the Innocet[®] B package
- TM/154/18 – B10 fuel with the Innocet[®] C package

Table 1. Physical and chemical properties of B10 diesel fuel

Property	Unit	Results
Cetane number	–	52.3 ±3.0
Cetane number derivative	–	53.41 ±0.18
Cetane index	–	55.0 ±0.8
Density at 15 °C	kg/m ³	830.4 ±0.4
PAH content	% (m/m)	1.1 ±0.5
Sulfur content	mg/kg	4.9 ±1.2
Manganese content	mg/l	< 0.5
Flash point	°C	60.5 ±2.5
Carbon residue (on 10% distillation residue)	% (m/m)	0.016 ±0.012
Water content	mg/kg	60 ±38
Total contamination	mg/kg	20.5 ±5.4
Cooper strip corrosion (3h at 50 °C)	corrosion rate	1
Oxidation stability (g/m ³)	g/m ³	8 ±7
Oxidation stability [110 °C] (Rancimat)	h	23.1
Lubricity, wear scar diameter at 60 °C	µm	165 ±73
Viscosity at 40°C	mm ² /s	2.730 ±0.022
Distillation		
% (V/V) recovered at 250 °C	% (V/V)	36.8 ±1.9
% (V/V) recovered at 350 °C	% (V/V)	93 ±1.9
95% (V/V) recovered at	°C	355.9 ±6.6
Fatty acid methyl ester (RME) content	% (V/V)	9.9 ±0.5
Ash content	% (m/m)	< 0.001

3. Characteristics of laboratory for engine tests

The engine tests of the fuels in question were performed at the Internal-Combustion Engines Laboratory of the Insti-

tute of Motor Vehicles and Internal-Combustion Engines of the Cracow University of Technology.

Tests for each of the tested fuels were carried out using a 4-cylinder diesel engine, type 1.9 TDI (AJM version), made by VW, whose technical characteristics are shown in Table 2.

Each of the 4 fuels made it possible to determine, on the background of the load characteristics of the engine (at the rotational speed corresponding to the maximum torque), 5 groups of engine parameters:

- energy parameters of engine (fuel consumption, total efficiency of diesel engine),
- composition of exhaust gases (concentration of CO, THC, NO_x and smoke),
- parameters of engine systems (e.g. start of fuel injection, temperature and pressure of air after turbocharger, temperature of exhaust gases, percent of EGR),
- parameters of indicator diagrams (e.g. start of self-combustion, maximum rate of combustion pressure, maximum of combustion pressure),
- parameters of heat release diagrams (e.g. maximum of kinetic and diffusion heat release, end of combustion).

Table 2. Technical parameters of VW 1.9 TDI (type AJM)

type of fuel ignition	self-ignition
cylinder layout	in-line
the number of cylinders	4
piston diameter	79.5 mm
piston stroke	95.5 mm
engine displacement	1,896 cm ³
geometric compression ratio	18
cylinders work order	1-3-4-2
rated gross power	85 kW
rated engine speed	4,000 rpm
maximum engine speed	4450 rpm
maximum engine torque	285 Nm
rot. speed of maximum torque	2,000 rpm
volume of the engine cooling system	6.1 dm ³
volume of the engine oil system	4.4 dm ³
lubricating oil pressure	0.11–0.41 MPa
fuel supply system	unit injectors
max. pressure of fuel injection	20 MPa

In order to measure and analyze the parameters of the engine fuelled sequentially by tested fuels was used part of the measuring systems, which are equipped laboratory of Cracow University of Technology. The study was carried out using the popular VW 1.9 TDI engine in the AJM version (with unit injectors), popular in Europe, in passenger cars and light commercial vehicles. This article describes the influence of engine load (at max torque speed 2000 rpm) on the basic engine parameters and exhaust composition. In the study were used Schenk engine dynamometer, AVL Fuel Balance (for measure of fuel consumption) and AVL Bench Emission System CEB II for measuring the concentration of gaseous components. In addition, the air temperature collected by the engine, the air temperature behind the turbocharger, behind the inter-cooler and into the intake manifold (behind the EGR valve) were measured. The exhaust gas temperature was measured before and after

the turbocharger. For comprehensive analysis, the impact of the engine load on the boost pressure and the start of fuel injection were measured. In the second part of the article, the AVL Indimeter 617D was also used to determine and analyze the fast-changing pressure in the engine cylinder (as a function of the engine crankshaft rotation angle) and to calculate the rate of heat release.

4. Experimental

4.1. Energy parameters of the engine

Because the impact of engine load on most of the analyzed parameters is much higher than the impact of the tested fuel additives, a linear load characteristics analysis is very difficult. Therefore, in order to perform a more accurate analysis, bar graphs were prepared, based on cross-sections of load characteristics, of all parameters for low, medium, and high load of the engine (50 Nm, 150 Nm, and 250 Nm). As a result, with changed ranges of the value of specific parameters (for different loads), it was easier to analyze the impact of the tested fuels on the analyzed parameter.

The characteristics of the fuels used in the tests are shown in item 2 of this paper.

The basic energy parameter of the engine (at a constant rotational speed and engine load) is hourly fuel consumption (G_{pal} [kg/h]) – Fig. 1.

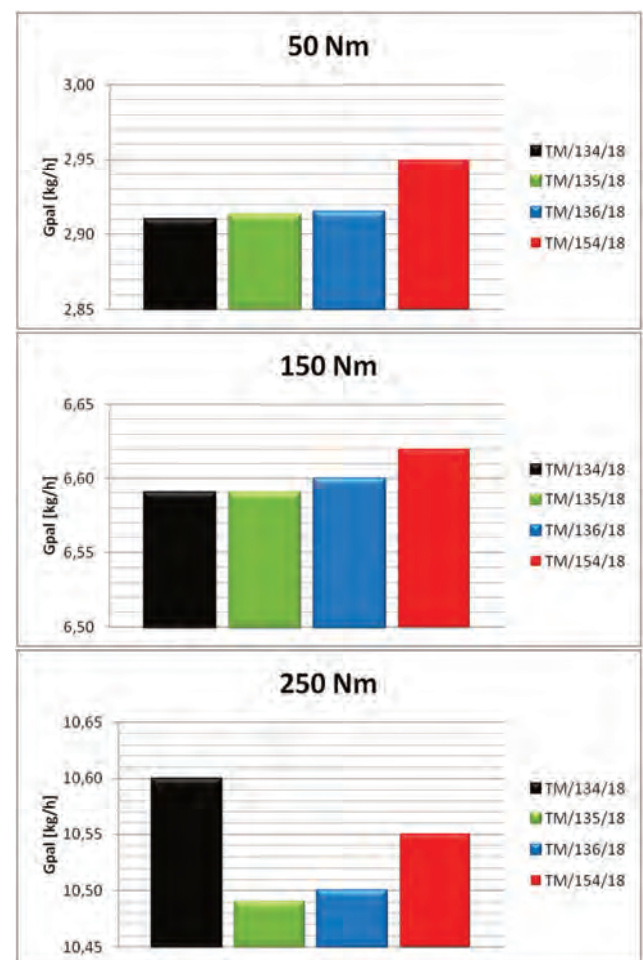


Fig. 1. Hourly fuel consumption in selected duty points of the engine for the four tested fuels

At low and medium engine load, no beneficial impact of the fuels with special additive packages (TM/135/18, TM/136/18, and TM/154/18) was identified, compared to the reference fuel (TM/134/18). The measured difference between the values G_{pal} in this range of engine load is equal to only approximately 1%. The situation is slightly different in the case of a high engine load when the hourly fuel consumption is slightly lower for all fuels with a special additive packages (TM/135/18, TM/136/18, and TM/154/18) compared to the TM/134/18 fuel. However, the change is equal to approximately 1%. The relevant additives in principle are supposed to influence the course of the combustion process, including the speed of kinetic and diffusive combustion in the engine chamber; consequently, their influence will be visible mostly in the composition of the exhaust gas, which is discussed in the next chapter.

Because the measurements were conducted at a constant rotational speed of the engine, at constant engine load the percentage changes of the hourly and unit fuel consumption and of the general efficiency of the engine must be the same. The conclusions of the analysis of the impact of engine load on the hourly and unit fuel consumption and on the general efficiency of the engine are certainly the same as the conclusions from the analysis of the hourly fuel consumption for the tested fuels.

4.2. Exhaust gas composition

From the point of view of the fast-changing processes in a piston combustion engine, as far as combustion gas composition is concerned, the most important are the combustion gas components that are limited during type testing. This group includes carbon oxide (CO), gaseous hydrocarbons (THC), nitrogen oxides (NO_x), particulate matter (PM), and non-toxic carbon dioxide (CO_2).

The principal objective of use of one of the additives in the tested fuels (TM/135/18, TM/136/18, TM/154/18) was to shorten the self-ignition delay t_{sp} and, consequently, to reduce the concentration of nitrogen oxides (NO_x) in the engine's exhaust gas. In the conditions present in the course of the combustion process in the cylinder of a diesel engine, emission of nitrogen oxides is related most of all to the maximum combustion temperature (to a smaller extent with availability of oxygen and duration of the nitrogen oxidation reaction). Consequently, the shorter the self-ignition delay (the shorter the time of gathering of the fuel dose in the combustion chamber until the self-ignition), the smaller the dose of fuel that undergoes self-ignition, which results in a less dynamic combustion process (lower maximum speed of kinetic combustion) and, consequently, lower maximum combustion temperature, which leads to a lower concentration and emission of NO_x in the exhaust gas.

On the other hand, a reduction of the self-ignition delay, t_{sp} , of the fuel leads to a reduction of the time of mixing of fuel with air before the fuel self-ignition, which results in incomplete combustion, which in turn increases the emission of particulate matter and smoke opacity of the exhaust gas. At the same time, a reduction of the t_{sp} shortens the time of gathering of fuel in the combustion chamber before self-ignition, which, as has been mentioned, reduces the maximum and average combustion temperature. A reduction of the combustion temperature leads to combustion of a

smaller quantity of soot and PM that have been produced. Consequently, all in all, a reduction of the self-ignition delay of the fuel involves an increased quantity of smoke opacity of the exhaust gas and an increased emission of particulate matter.

Based on the above brief analysis of the mechanisms of occurrence of emission of particulate matter (PM) and nitrogen oxides (NO_x), research on various diesel engines tested for years in laboratories worldwide has demonstrated that use of one design or regulation or fuel method, which reduces the self-ignition delay, reduces the emission of NO_x but, at the same time, increases the emission of PM. This is shown schematically in Fig. 2.

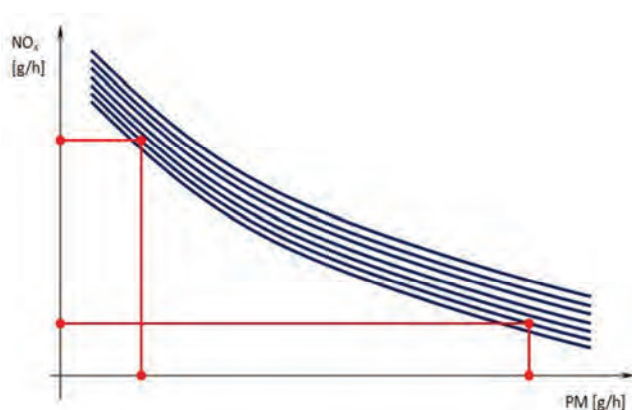


Fig. 2. Diagram of correlation between NO_x and PM emissions

If design and regulation methods are used for the engine or for fuel additives, usually several procedures are performed that simultaneously reduce the emission of NO_x and PM (including smoke opacity D) in exhaust gas from a diesel engine.

Figure 3 shows the impact of engine load on concentration of nitrogen oxides (NO_x) in exhaust gas of an engine supplied, consecutively, with the four tested fuels TM/134/18, TM/135/18, TM/136/18, and TM/154/18. The data shown in Fig. 3 indicates that at a low engine load use of fuels with a special additive package does not have a significant impact on the concentration of NO_x in the exhaust gas. The observed differences between the concentration of NO_x for the fuels used are equal to just several ppm and are within the range of repeatability of the operation of the engine and the measurement accuracy. In the quantitative sense, the situation is similar for the average engine load; however, in this operation range the concentration of nitrogen oxides in the exhaust gas is systematically only a little lower for fuels with the special additives packet (TM/135/18, TM/136/18, and TM/154/18) compared to the reference fuel (TM/134/18). Only at high engine loads, the impact of one of the fuel additives, TM/154/18, on a reduction of concentration of nitrogen oxides in the exhaust gas compared to other fuels, especially TM/134/18, becomes clearly visible. The difference is 26 ppm (3.2%) at the load of 250 Nm. For the maximum engine load that has been applied (280 Nm), the difference is as large as 7%.

The intensity of the impact of this type of fuel additives quite often depends on the engine temperature and, consequently, on the engine load. A similar phenomenon is also

observed when comparing the concentration of NO_x in exhaust gas of an engine supplied with fuels TM/134/18 and TM/154/18. Only at high engine loads, the concentration of nitrogen oxides in the exhaust gas is significantly lower for the TM/154/18 fuel than for the TM/134/18 fuel.

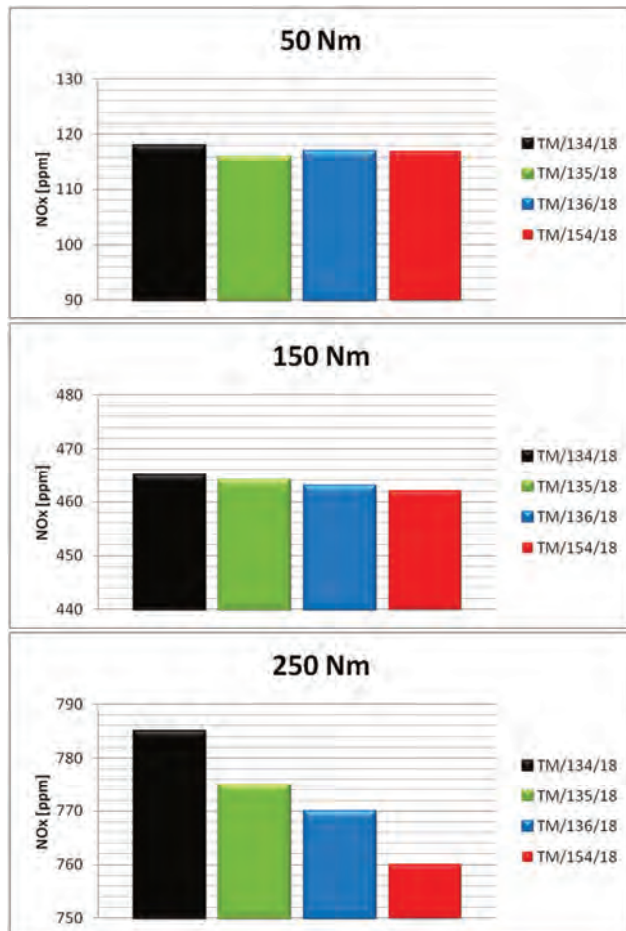


Fig. 3. NO_x concentration in exhaust gas in selected duty points of the engine for the four tested fuels

Due to the adopted test schedule, emission of particulate matter (PM) was not measured. However, because emission of particulate matter is largely related qualitatively to the smoke opacity of exhaust gas (D), the latter parameter was measured using the Bosch filtration method. The smoke opacity (D) for various loads and for the analyzed fuels is shown in Fig. 4. The accuracy of the measurement of smoke opacity (D) using the Bosch filtration method is equal to 0.05 °B. Thus, one can see that at low and medium engine loads, the fuels used do not have a significant impact on the smoke opacity of the exhaust gas; consequently, they most likely also do not have a significant impact on emission of particulate matter (PM). Qualitatively, the exception is the TM/154/18 fuel, which at all engine loads is characterized by the highest smoke opacity of the exhaust gas. This is particularly evident at high engine load where the difference in the smoke opacity of the exhaust gas of the engine supplied with the comparative fuel TM/134/18 and the fuel TM/154/18 is as much as 0.15°B – Fig. 4. At the same time, one must keep in mind that at that duty point of the engine, that fuel is characterized by the lowest concen-

tration of NO_x in the exhaust gas – Fig. 3. Consequently, the relationship shown schematically in Fig. 2 is confirmed again – the engine or fuel method that reduces the concentration of NO_x in the exhaust gas also increases the smoke opacity of the exhaust gas. What was also confirmed is that the intensity of impact of the fuel additives usually was greater at higher engine loads.

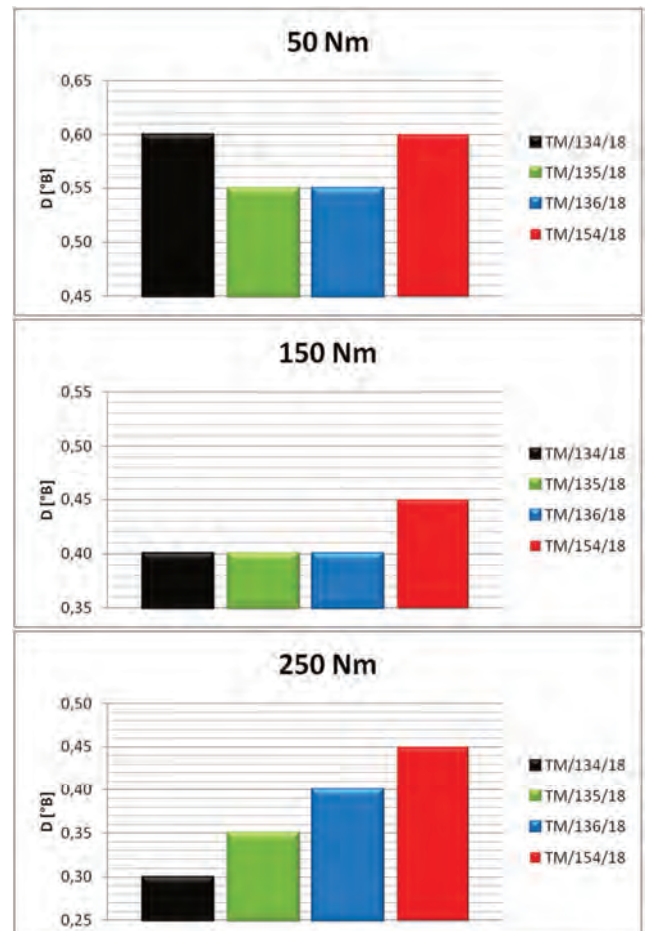


Fig. 4. Smoke opacity of the exhaust gas (D) at selected duty points of the engine for the four tested fuels

Gaseous hydrocarbons (THC) in exhaust gas of a diesel engine are the result of incomplete combustion and their formation, even at sufficient quantities of oxygen, is associated mostly with too low temperature in the fuel combustion zone. Because contemporary diesel engines, including the engine used in the tests, due to the use of various design and regulation measures, are characterized by very low concentrations of THC in exhaust gas (even upstream of the oxidizing catalytic converter) and their concentration is usually equal to several or up to twenty ppm, this component was intentionally not measured during the tests. Moreover, the conversion efficiency of commonly used oxidizing catalytic converters is so high that in reality the concentration of THC in the exhaust gas emitted from diesel engines is insignificant.

From a chemical standpoint, formation of carbon oxide (CO) in combustion conditions in the cylinder of a diesel engine is related to the global shortage of oxygen (homogeneous mixture). On the other hand, in a cylinder of a diesel

engine, formation of CO (even given the global shortage of oxygen) is possible due to local shortage of oxygen around the jets of injected fuel after its self-ignition and in the combustion process (heterogeneous mixture).

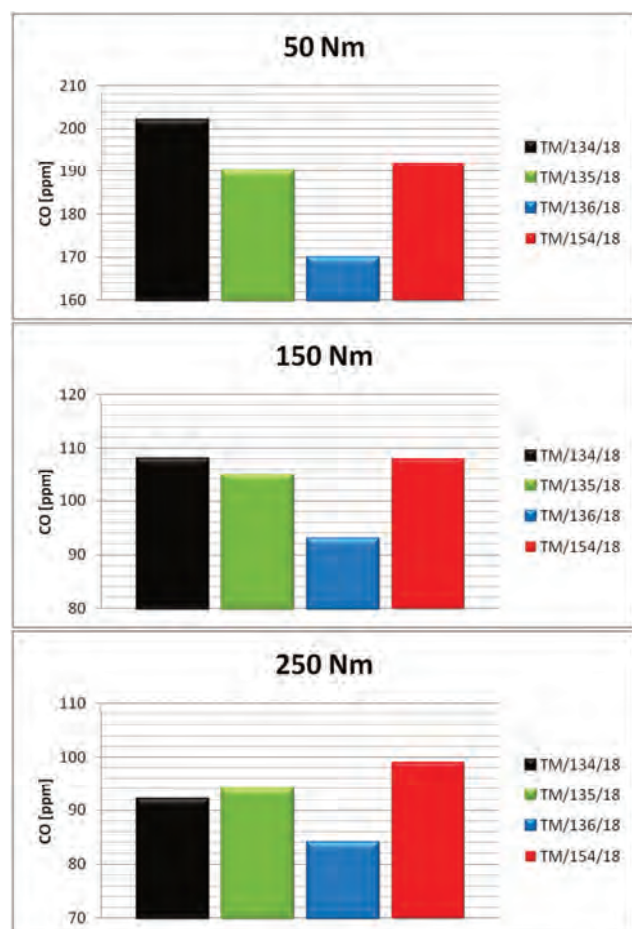


Fig. 5. CO concentration in exhaust gas in selected duty points of the engine for the four tested fuels

The concentration of carbon oxide (CO) presented for the selected fuels in Fig. 5 in the entire range of load of the diesel engine is definitely smaller than in spark-ignition engines, which is due to the range of the value of the air excess λ used in diesel engines. Nevertheless, the differences in the CO concentrations measured in the exhaust gas for the engine used in the tests and for the analyzed fuels are very small. Moreover, as mentioned in the first paragraphs of the chapter, the catalytic (oxidizing) converters that are used make it possible to eliminate those differences. Consequently, in practice, the concentration of THC and CO upstream of a catalytic converter is not important.

As mentioned in the first paragraphs of this chapter, the concentration of CO₂ in the exhaust gas is correlated with the quantity of consumed fuel. However, because the fuel additives affect the hourly consumption of fuel only to a minimum extent (within the limits of repeatability of the operation of the engine – approx. 1%), then also the concentration of carbon dioxide (CO₂) – Fig. 6 – in the exhaust gas of the engine, for the analyzed fuels, shows little difference regardless of the engine load.

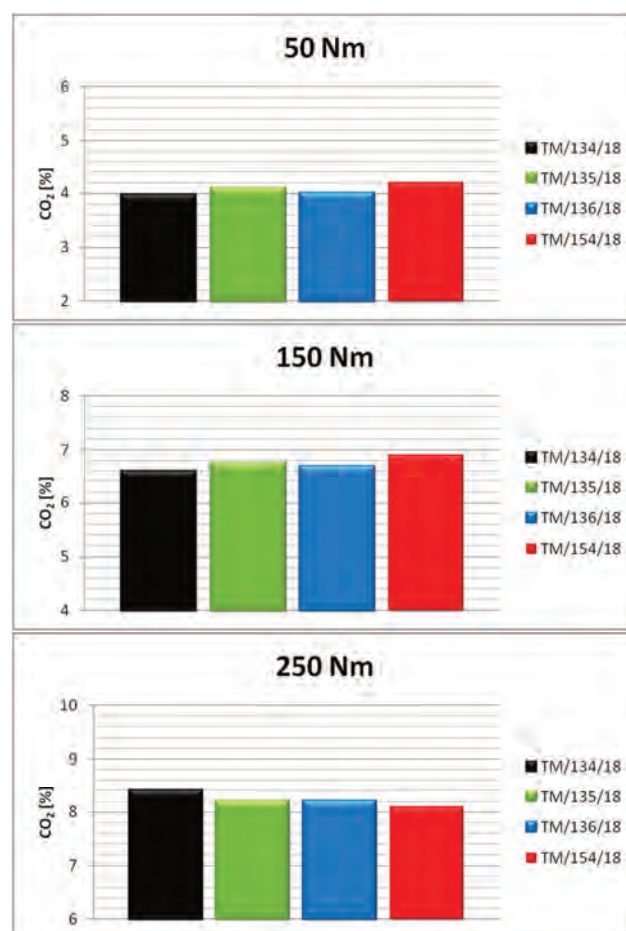


Fig. 6. CO₂ concentration in exhaust gas in selected duty points of the engine for the four tested fuels

In order to better understand the impact of the tested fuels on the composition of the exhaust gas, an analysis was performed of the indicator diagrams and the heat release speed in the cylinder of the engine.

In order to better understand the impact of the tested fuels on the composition of the exhaust gas and in order to perform a cause and effect analysis of the occurring phenomena, the measurements were expanded to include indication of the engine cylinder and calculations of the change of the heat release speed as a function of the angle of rotation of the engine crankshaft. A synthesis of the measured parameters is presented in the second article by the same authors, titled "Influence of cetane-detergent additives into diesel fuel with RME content increased to 10% on the parameters of indicator diagrams and rate of heat release in a diesel engine."

Conclusions

- The tests that were performed made it possible to draw the following major conclusions concerning a comparison of the impact of the tested fuels (TM/135/18, TM/136/18, and TM/154/18), compared to the reference fuel (TM/134/18):
- The tested fuels (TM/135/18, TM/136/18, and TM/154/18) have no practical impact (compared to the TM/134/18 fuel) on the hourly fuel consumption (and, consequently, also on the overall efficiency of the en-

gine). The only differences observed during the tests are within the limit of approx. 1%, which is within the limit of engine operation repeatability.

- The TM/135/18 fuel has an impact on the engine operation process basically only at low engine loads. At such loads, a small reduction of the concentration of nitrogen oxides (NO_x) in the exhaust gas of the engine was observed.
- The TM/136/18 fuel has a different qualitative action compared to the TM/135/18 fuel because its impact on the operating process of the engine is noticeable only at medium and high engine loads. At such loads, the concentration of NO_x in the flue gas of the engine was reduced (compared to the comparative fuel TM/134/18). Because usually a reduction of NO_x concentration in exhaust gas is accompanied by an increase of smoke opacity of the exhaust gas, also in this case the TM/136/18 fuel was characterized by a slightly higher smoke opacity of the exhaust gas at high engine loads (compared to the TM/134/18 fuel). Due to the small practical impact of the TM/136/18 fuel on a reduction of NO_x concentration in the exhaust gas, a decision was made to use the TM/154/18 in which the concentration of the additive that shortens the τ_{sp} is twice as high as the concentration in the TM/136/18 fuel.
- The TM/154/18 fuel is characterized by similar qualitative changes of the analyzed operating parameters of the

engine as the TM/136/18 fuel, whereby the changes are much greater at medium and high engine loads. The impact of that fuel at lower engine loads (0–80 Nm), with reference to the energy parameters and the composition of the exhaust gas of the engine, is negative. Only at higher loads (up to the maximum load), the additives used in that fuel start to have a positive impact on a reduction of NO_x concentration in the exhaust gas of the engine (compared to the TM/134/18 fuel). Unfortunately, in those conditions there is also an increase in the smoke opacity of the exhaust gas.

- In order to perform more in-depth analyses of the impact of the used fuel additives on the operating parameters and the composition of the exhaust gas of the engine and to determine the appropriate direction of further quantitative and qualitative changes in the fuel additive packages, one must conduct an evaluation of the engine operating process. For this purpose, the second article, titled "Influence of cetane-detergent additives in diesel fuel with RME content increased to 10% on the parameters of indicator diagrams and rate of heat release in a diesel engine," presents an analysis of the heat release speed for the tested fuel.

Acknowledgements

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Nomenclature

CI compression ignition
CO concentration of carbon monoxide
D smoke of exhaust gases
Gpal hourly fuel consumption

HC concentration of hydrocarbons
 NO_x concentration of nitrogen oxides
RME rapeseed methyl ester
PAH polycyclic aromatic hydrocarbons

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Application of the experimental design technique in fuel dose adjustment of common rail injector

The article presents a simplified methodology for testing Denso common rail electromagnetic injectors, which was based on an adjustment of idling fuel dosage. In order to determine the parameters significantly affecting this process, the design of experiments technique (DoE) was applied. Due to the non-linear course of the rates characteristics obtained on the diagnostic test bench, the trivalent fractional plan was used. Analysis of computational data has shown that the generated parabolic model is well suited to the actual measurement results, as the injector dosage is slightly out of projected values. The calculations were carried out in the STATISTICA computer program, which takes into account the specificity of the experimental theory methods and is one of the most universal tools available on the market. Innovation is the effective implementation of the proposed methodology in the repair process, which has not been taken into account in this regard yet.

Key words: common rail injector, design of experiments technique, fuel dose adjustment, idling

1. Introduction

One of the most common malfunctions in common rail injectors is the deterioration of the cooperation between the control needle and the spray nozzle [7]. It may be the result of normal or accelerated use of the precision pair. In the first case, the fault usually becomes apparent after a longer period of engine operation [9]. However, pathological wear takes place in a much shorter time, usually due to insufficient lubrication [21], overheating of the contact zones [1] and chemical influence of combustion products [8]. Since the discussed parts are in direct contact with the fuel supplied, the decisive factor is its quality [14], physicochemical properties and the presence of solid [17] and chemical contaminants [3]. The friction wear of the precision pair affects the injector dosage processing an unfavourable way as it results in an increased flow of return flows, i.e. the amount of fuel reaching the overflow. As a result, idling and low load operation becomes uneven. After disassembling the injector, the fault can be detected at the low operating pressure on the diagnostic test bench. However, in the other areas of work, the correct values of fuel injection rates are observed because the lift of the control needle is not hindered and it overcomes the resistance of the needle spring without any obstacles. In such situations, the test procedure can be simplified by eliminating the aforementioned malfunction by adjusting the idle dose.

Denso electromagnetic injectors are characterized by their construction and operating principle, similar to Bosch products of the first generation [11]. For this reason, their regulation is carried out in the same way. Unfortunately, until now the manufacturer has not released a comprehensive repair technology, so the impact of construction parameters on the value of individual rates remains difficult to estimate [12]. With regard to the idling speed, the change of the factory settings is obtained by replacing the needle and/or calibration shims. Reducing the thickness of the first one changes the preload of the lower spring, which stops the height of the control needle and consequently the amount of fuel injected. In turn, the second one, located between the electromagnet coil and the main body, affects

the overall injector dosage (Fig. 1). Therefore, the increase in its thickness may be carried out to a limited extent [13].

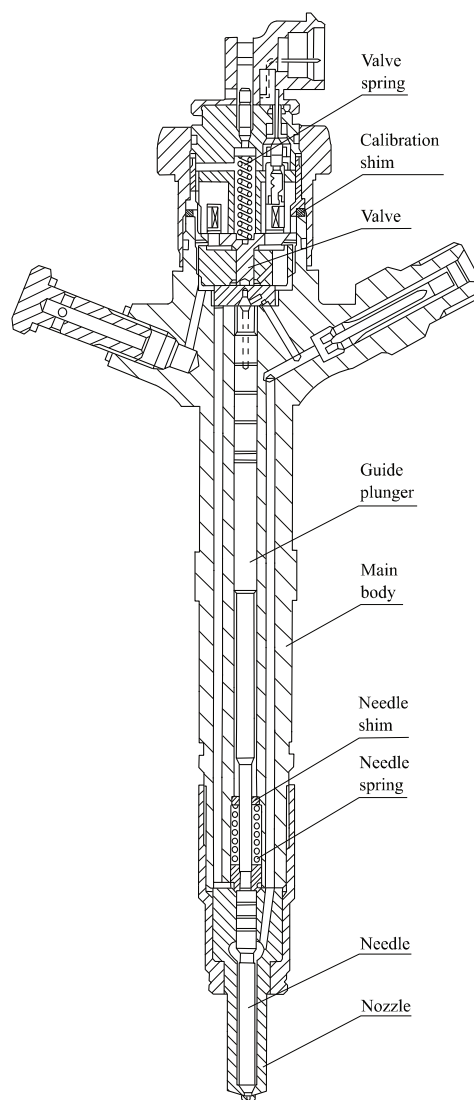


Fig. 1. Denso injector construction

In order to determine the parameters that have a significant impact on the idling rate, the design of experiments technique was used, which in this respect has not been taken into account until now. The application of this methodology allowed to solve the problem of unambiguous determination of the dimensions of regulatory shims, and thus their precise selection out of specific selection groups. As a result, the testing process of Denso electromagnetic injectors on diagnostic test benches has been simplified.

2. Methods

2.1. Test object and test beds

The test object was a Denso electromagnetic injector, dismounted from a 2.2 HDI (*High pressure Direct Injection*) engine of a Citroën Jumper II delivery van with an operational mileage of 156 thousand km. It is a four-cylinder, 16-valve drive unit with direct fuel injection in the common rail system.

The research was carried out according to the adopted own methodology, including two stages of implementation (Fig. 2).

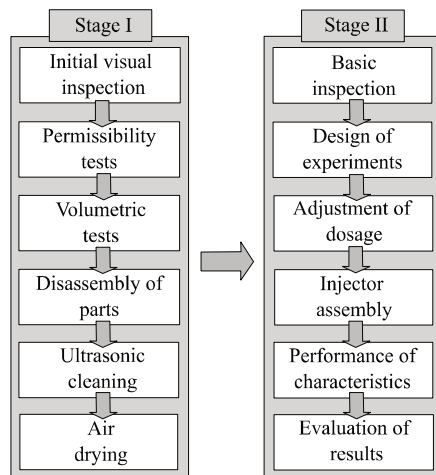


Fig. 2. Denso injector research methodology

The first of these began with the external examination of the injector, as well as the admission measurements, which included the following types of tests: electric – eRLC (*Resistance, Inductance, Capacitance*), LKT (*Leak Test*) and NOP (*Nozzle Opening Pressure*). In this phase, an auxiliary Bench CRU 2 Zapp diesel diagnostic test bench was used (Fig. 3). Volumetric tests iVM (*Injector Volume Metering*) were also conducted at the same station. They were carried out at various pressures and control times for doses of: full load VL (Ger. *Vollast*), emission EM (Ger. *Emissionspunkt*), idling LL (Ger. *Leerlauf*) and pilot VE (Ger. *Voreinspritzung*). Then the injector was disassembled into components, cleaned in ultrasonic washers and dried with compressed air.

In the second stage, the most important control and regulation parts were inspected using a laboratory microscope. Prior to assembly, the injector underwent a dose correction, which was based on the results of statistical analysis calculations. Idle characteristics were based on experimental data that was obtained on the STPiW-3 diagnostic test bench (Fig. 4).

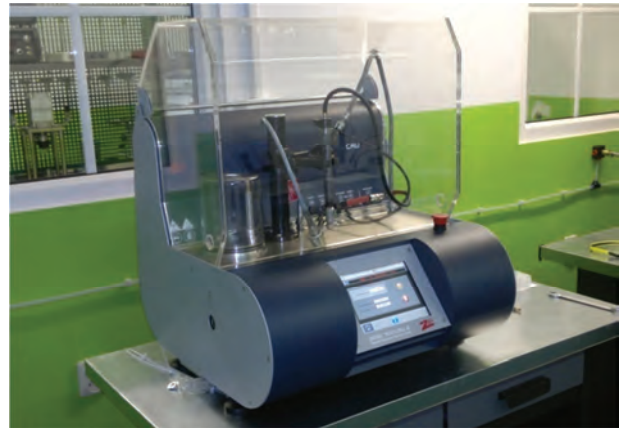


Fig. 3. Diesel Bench CRU 2 Zapp auxiliary diagnostic test bench



Fig. 4. Diesel Bench STPiW-3 main diagnostic test bench

2.2. Preliminary tests results

Electrical measurements excluded the failure of the electromagnet coil and its windings were not faulted to frame. The nozzle opening pressure was also positively evaluated and no leakage of test liquid was found in the leak test (Table 1).

Table 1. Approval tests results

Test type		Result
Electric test eRLC	Coil resistance, R_c [Ω]	0.72
	Coil inductance, L [μH]	180
	Winding faulted to frame	Not found
Leak Test LKT [MPa]		85.80
Nozzle Opening Pressure test NOP [MPa]		16.50

Table 2. Volumetric tests results „iVM”

Dose type	Injection pressure, p_{inj} [MPa]	Duration of the electrical pulse (control), t [μs]	Nominal and actual rate, D [$\text{mm}^3/\text{injection}$]
Fullload VL	160	910	57.99 ± 8.41 58.69
Emission EM	80	720	22.18 ± 5.10 21.13
Idling LL	25	710	4.73 ± 2.23 2.02
Pilot VE	80	320	2.23 ± 1.48 0.91

Analysis of the results obtained in the volume tests showed that the injector is working properly. The exception was the too low idle dose value, which did not fall within the limits set by the manufacturer (Table 2).

2.3. Design of experiments

According to the design of experiments theory, one of the basic and most important tasks is to determine a set of quantities describing the object under study. Their number can be limited by thoughtful and careful allocation to particular groups [19]. This allows to obtain reliable results on the basis of a relatively small number of observations [2]. In this way, the labour consumption of conducted activities is reduced and the need to consider a comprehensive analysis of a given process is avoided [16].

In the present case, it was arbitrarily assumed that the idling period (dependent variable) is influenced by the following input quantities (independent variables) such as: time of control t , thickness of the needle shim T_{ns} and the calibration shim T_{cs} . It was also assumed that the interaction of disturbances and the constant factor is negligibly small, which means that they may not be included in further considerations (Fig. 5). All measurements were carried out in similar ambient conditions, at the minimum injection pressure ($p_{inj} = 25$ MPa).

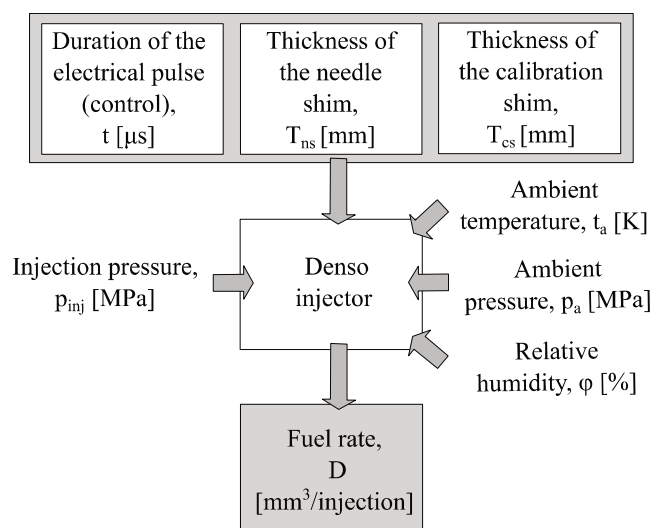


Fig. 5. Qualities defining the object under research

Conducting an experiment requires establishing the methodology that determines the selection of the correct plan [6]. It should guarantee a high rationality of research and enable obtaining the most reliable information about the object. In most cases, however, selective data, that is relevant to a specific problem, is sufficient [5]. For this reason, the determined plans are often chosen as they guarantee the ease of the mathematical preparation of the results [20]. At the same time, they retain the ability to create approximating functions reflecting the physical aspect of the experiment, while limiting the number of measurements required in their implementation [15]. In addition, the course of characteristics obtained on the STPiW-3 diagnostic test bench indicates that the injector's rate is described by a non-linear relationship [10]. For this reason, it was

decided to use a trivalent fractional plan $3^{(k-s)}$. This means that the input quantities in the number k assume values on three levels of variation ($-1, 0, 1$), where s is the degree of bisection [4].

Based on the assumptions made, which are related to the characteristics of the tested object, the selection plan $3/1/9$ was created in the STATISTICA program. This record indicates that for 3 independent variables called generators (X, Y, Z), the tests will be conducted in 1 block and 9 measuring points (Table 3).

Table 3. Trivalent fractional plan $3^{(k-s)}$

Point number	Independent variables (generators)		
	X	Y	Z
1	-1	-1	-1
2	-1	0	1
3	-1	1	0
4	0	-1	1
5	0	0	0
6	0	1	-1
7	1	-1	0
8	1	0	-1
9	1	1	1

In the next step, the names of the input quantities and the corresponding numerical values were introduced (Table 4).

Table 4. Selection of input quantities in real space

Generator	Input quantity (generator)	Value		
		Lower	Central	Upper
X	Duration of the electrical pulse (control), t [μ s]	200	700	1200
Y	Thickness of the needle shim, T_{ns} [mm]	1.26	1.30	1.34*
Z	Thickness of the calibration shim, T_{cs} [mm]	1.60*	1.64	1.68

* data obtained by measurement

They were determined based on thickness measurements of shims using a standard digital micrometre. A selection of their new alternatives from well-defined selection groups was also made. In this aspect, the need to increase idle rate as well as the central position of the central plan value, which is the arithmetic mean of the lower and upper stellar points, was taken into account. The duration of electrical pulses, i.e. the control of the injector, was selected in the same way.

3. Analysis results and discussion

Table 5 presents the final plan of the experiment, the implementation of which can be carried out according to the variable order of individual assemblies. It should be noted that every replacement of the shims requires disassembly and re-assembly of the injector, which unnecessarily extends the time of its adjustment.

The statistical analysis also included squares of independent variables, which were introduced to estimate the

Table 5. Trivalent fractional plan $3^{(k-s)}$ in the STATISTICA worksheet

Point number	t	T_{ns}	T_{cs}	t^2	T_{ns}^2	T_{cs}^2	D
1	200	1.26	1.60	40000	1.5876	2.56	0.2
2	200	1.30	1.68	40000	1.69	2.8224	0.1
3	200	1.34	1.64	40000	1.7956	2.6896	0.1
4	700	1.26	1.68	490000	1.5876	2.8224	3.2
5	700	1.30	1.64	490000	1.69	2.6896	2.8
6	700	1.34	1.60	490000	1.7956	2.56	1.9
7	1200	1.26	1.64	1440000	1.5876	2.6896	7.1
8	1200	1.30	1.60	1440000	1.69	2.56	6.2
9	1200	1.34	1.68	1440000	1.7956	2.8224	5.9

Table 6. Summary of multiple regression in the STATISTICA program

Number of plan points: 9	Correlation coefficient $R = 0.99947492$; Determination factor $R^2 = 0.99895011$; Fisher-Snedecor test statistics $F(5.3) = 570.89$; Severity level $p < 0.00012$; Estimation standard error: 0.4657					
	Regression predictor, b^*	Standard error with b^*	Coefficient of function, b_f	Standard error with b_f	b_f quotient and its standard error, $t_i(3)$	Severity level, p
Free term	–	–	–692.7175556	174.1738353	–3.97716	0.028431
T	0.4252	0.092634	0.0027200	0.0005926	4.59010	0.019425
T_{ns}	–0.1355	0.018707	–10.8333333	1.4958791	–7.24212	0.005430
T_{cs}	10.7290	2.657028	857.9166667	212.4622334	4.03797	0.027324
t^2	0.5661	0.092634	0.0000025	0.0000004	6.11104	0.008806
T_{cs}^2	–10.6824	2.657028	–260.4166667	64.7734656	–4.02042	0.027637

coefficients of the approximating function to a non-linear form. In turn, the final column shows the final results of idle rate. In the case of multiple regression, the approximating function is obtained in such a way as to minimize the difference between the modelled value and the actual dependent variable for the input quantities.

The data summarized in Table 6 shows that for the considered case it will be at most a polynomial of the second degree, which, after taking into account the coefficients b_f , will take the form:

$$D = -692.7 + 2.7 \cdot 10^{-2} \cdot t - 10.8 \cdot T_{ns} + 857.9 \cdot T_{cs} + 2.5 \cdot 10^{-6} \cdot t^2 - 260.4 \cdot T_{cs}^2 \quad (1)$$

Analysis of computational data indicates that the generated parabolic model is very well suited to the actual results of the experiment. The qualitative measures are the coefficients of: multiple correlation R and determination R^2 , which are close to unity. The proper selection of the function is also confirmed by the high measure of the Fisher-Snedecor test and the corresponding level of significance ($p < 0.00012$). In addition, the standard error of the estimation allows to conclude that the actual idling rate is slightly out of the projected value.

From the statistical point of view, almost all input quantities were significant and therefore they were included in the regression function. However, the lowest value of significance level was obtained for the thickness of the needle shim, which had a decisive influence on the injector rate. It follows that the idling adjustment should be limited only to reducing the preload of the needle spring. It should bring the expected results in a simplified research methodology. In turn, the replacement of the calibration shim is not recommended, because this way of regulation will have an

indirect impact on the value of other rates and will force the need to check them in additional tests.

Figure 6 shows the characteristics of the tested injector. Thanks to the regulation, an increase in idle running rates was obtained, which were within the limits set by the manufacturer. It can also be seen that the waveforms of the approximation curves slightly deviate from the actual measurement results. For example, for the control time $t = 200 \mu s$ and the simultaneous exchange of both shims, the estimated dose value was $0.7 \text{ mm}^3/\text{injection}$, while on the diagnostic test bench the value of $0.4 \text{ mm}^3/\text{injection}$ was obtained (Table 7).

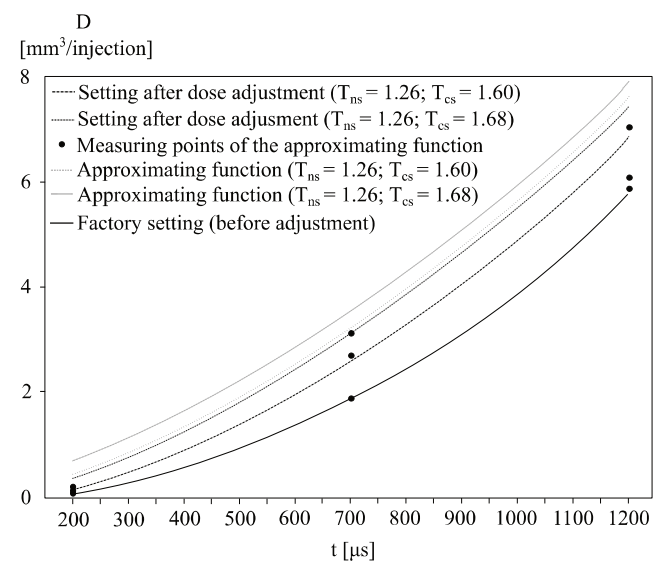


Fig. 6. Idle characteristics of the tested injector

Table 7. Final results of tests and calculations

Duration of the electrical pulse (control), t [μ s]	Factory setting	Setting after dose adjustment			
	$T_{ns} = 1.34$ $T_{cs} = 1.60$	$T_{ns} = 1.26$ $T_{cs} = 1.60$		$T_{ns} = 1.26$ $T_{cs} = 1.68$	
200	0.1	0.2	0.4*	0.4	0.7*
450	0.8	1.2	1.7*	1.6	2.0*
700	1.9	2.6	3.3*	3.2	3.6*
950	3.6	4.5	5.3*	5.2	5.6*
1200	5.7	6.9	7.6*	7.5	7.9*
* data estimated analytically					

The data presented in the co-author's publication [18] shows that the repair of Denso electromagnetic injectors is characterized by very high efficiency with relatively small operational runs. In most cases, this process requires adjustment of idle dosage, which justifies the desirability of implementing the proposed calculation methodology in a wider scope. Nevertheless, over 200,000 km it is recommended to replace the precision pair with a new one (needle-spray nozzle), regardless of the type of failure identified. However, the growing number of failures of other regulation and control elements remains a problem. Their removal is not always possible, mainly due to limited access to original spare parts, which in turn is the manufacturer's conscious policy. As a result, the adoption of injectors coming from diesel engines with significant mileage is limited, because their repair becomes unreasonable or difficult to implement.

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gures will be printed as delivered. Only in very few exceptional cases will other forms be accepted.

4. Conclusions

The design of experiments technique (DoE) allowed to determine the influence of input quantities on the object's response in a precise way. In this respect, the choice of the trivalent fractional plan should be assessed positively because the number of measurements carried out has been limited to the necessary minimum, while meeting the requirements for their implementation. An undoubted advantage of approximation is the possibility of good approximation with a low polynomial degree, which further simplifies the way of calculations. It has been shown that the implementation of factor experiments provides reliable quantitative data that can be used to support the repair process, as illustrated by the example of the Denso electromagnetic injector. It is worth noting that in the case of comprehensive research, all possible combinations of independent variables should be considered ($3^3 = 27$ points). Of course, such a solution provides more information, but it is much more labour-intensive and seems to be economically unjustified.

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The repair of injectors was conducted on laboratory stands at VASCO Co. Ltd in Mierzyn near Szczecin, cooperating with the Department of Automotive Engineering of the West Pomeranian University of Technology (ZUT) in Szczecin.

Nomenclature

b^*	regression predictor	p_{inj}	injection pressure
b_f	coefficient of function	R	correlation coefficient
D	rate	R^2	determination factor
DoE	design of experiments technique	R_c	coil resistance
EM	emission dose (Ger. <i>Emissionspunkt</i>)	s	degree of bisection
eRLC	electric test (Resistance, Inductance, Capacitance)	t	duration of the electrical pulse (control)
F	Fisher-Snedecor test statistics	t_a	ambient temperature
HDI	High pressure Direct Injection	t_i	b_f quotient and its standard error
iVM	volumetric tests (Injector Volume Metering)	T_{cs}	thickness of the calibration shim
k	number of input quantities	T_{ns}	thickness of the needle shim
L	coil inductance	VE	pilot dose (Ger. <i>Voreinspritzung</i>)
LKT	Leak Test	VL	full load dose (Ger. <i>Vollast</i>)
LL	idling dose (Ger. <i>Leerlauf</i>)	X	independent variable (generator)
NOP	Nozzle Opening Pressure test	Y	independent variable (generator)
p	severity level	Z	independent variable (generator)
p_a	ambient pressure	φ	relative humidity

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Experimental study of carbon nanotubes in high viscosity lubricants

Carbon nanotubes (CNTs) beneficial tribological properties were reported in numerous studies performed mostly in idealized tribometer conditions, previously described full engine tests confirmed that CNTs added to the lubricating oil reduced friction significantly. Apparently the engine application requires low CNT concentration in oil, otherwise when increasing the CNT concentration beyond a certain level oil viscosity increases, rapidly forming a greasy substance not suitable for engine lubrication. In this paper we report a study of tribological properties of such a high viscosity lubricant performed in Amsler tribometer in comparison to high quality commercially available lithium grease. The CNT based high viscosity lubricant in its current formulation turned out as not reliable for general application due to its low adhesion to the metal surface. In contrary to some samples made of a polymer widely used in industry as a friction component, we observed wear reduction reaching nearly 50% when replacing the best suitable lithium grease by a CNT based lubricant. This effect was first observed after 30 min into a friction test, suggesting a CNT related mechanism of anti-wear protection which still need to be explained.

Key words: carbon nanotubes, friction, grease

1. Introduction

Carbon Nanotubes (CNTs) 're-discovered' by Sumio Iijima in 1991 have been intensively studied in the recent years; their often-reported unique combination of excellent physicochemical properties is extremely beneficial for tribological performance [1–4]. Despite the promising preliminary laboratory test results presented in the literature the early attempts of CNTs commercialization were disappointing [5]. In our opinion there are two main aspects that must be addressed before attempting large-scale industrial application of CNTs in lubricants. These aspects are: the stability of CNTs dispersion, and understanding of the mechanisms of CNTs tribological effects. Our engine tests described previously [6, 7] lead to the conclusion, that only well dispersed CNTs contribute to any significant friction reduction. The same level of total engine friction reduction, reaching a notable value of about 5%, was measured both for CNTs concentration in oil equal to 0.5% and in the case for concentration over ten times lower. In the first case only standard ultrasound homogenization was used, but a major improvement in the CNTs dispersion was achieved for the low CNT concentration using a simple and effective chemical methods. In [7] a smart solution was presented for constant CNT release into the lubricating oil, leading to a perfect dispersion and an unprecedented friction reduction of up to 16%. On the other hand our early attempts to exploit the friction reducing ability of CNTs by increasing their concentration in oil seemed to fail because at concentration as high as 1% the oil was converted during engine operation into a greasy substance that became too viscous to pump [6]. Disappointing engine test result turned into a new fascinating project where the high viscosity lubricant taken directly from the engine (labeled S0 in experiments described in this paper) was tested outside the engine and compared with top quality grease. A set of relevant tribometer test results was presented in this paper in an effort to achieve a detailed understanding of the physical phenomena related to the CNT effects in lubricants. Depending on the actual local friction conditions CNTs may reveal various

functional mechanisms occurring simultaneously during the friction process [8–10].

In this study we used Amsler tribometer with rotating steel ring, pushing against the samples made of cast iron, brass, aluminum and polymer, forming a friction pair lubricated with CNT based lubricants of various formulation and high quality commercially available grease as a reference sample. For our tests we used a polymer widely used as a material for friction components due to its low cost, good mechanical properties, low friction coefficient, and easy manufacturing. Until now the knowledge concerning manufacturing and tribological properties of CNT based high viscosity lubricants is marginal and partly incoherent [11, 12].

2. Materials and methods

2.1. Lubricants

High viscosity lubricants based on carbon nanotubes as thickener were tested in a tribometer and compared to standard lithium grease as commercially available top quality product based on PAO synthetic oil thickened with lithium soap (S1). All lubricants tested in the trials described below can be found in the Table 1.

Table 1 Formulation of lubricants

Lubricant	OIL	Thickener
S0	Mobil 1 0W/40	1% MWCNTs
S1	Synthetic, PAO + additives	Lithium complex
S2	Synthetic, PAO + additives	Lithium complex 0.5% MWCNTs short
S4	Orlen Platinum 5W/30	3% MWCNTs short
S5	VW Oil 5W/30	2% MWCNTs short ¹
S6	Castrol RS 10W/60	1% MWCNTs short ¹ 0.2% MWCNTs long ² 0.2% Graphene

¹ outer diameter 9.5 nm, length 1.5 μ m

² outer diameter > 50 nm, length 10...20 μ m

The S0 lubricant was produced accidentally in engine tests described in [6], but the lubricants S4, S5, S6 were obtained in

ultrasound disintegrator. Lubricant S2 is standard S1 grease mixed with CNTs.

2.2. Tribometer

The friction tests were carried out with the use of equipment located in the Sustainability and Wear Laboratory at the Poznan University of Technology (Institute of Machines and Motor Vehicles).

For comparative tests, the Amsler A-135 friction machine was used. The friction node consisted of a rotating cylindrical ring, against which a rectangular sample was pressed facing the friction surface. The rings in the tests has an outer diameter of 40 mm and a thickness of 10 mm, and were made of hardened steel with a hardness of 60 ± 2 HRC on the friction surface. The samples were rectangular $15 \times 10 \times 5$ mm elements made of various metal and polymer materials.

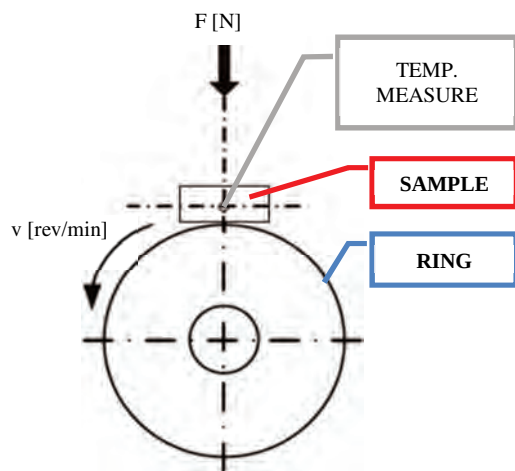


Fig. 1. Kinematics of Amsler Tribometer with tested sample pressed under controlled load against rotating ring

Before and after the tests, the tribometer performed surface profilograms of the samples and rings using a ZEISS contact profilometer equipped with heads with an inductive transducer and SAJD METROLOGIA software called SUFORM allowing the measurement and analysis of straightness and surface roughness deviations. The measurements were carried out using a measuring arm with diamond contact tip (without glider) in the shape of a pyramid with a rounded tip. The vertical range for measurements using the induction head was $\pm 200 \mu\text{m}$. The speed of the measuring tip travel used during measurements was 0.1 mm/s (the device's capabilities are 0.01 mm/s to 1 mm/s). The measurements were made by setting a single segment size to be $\lambda_c = 0.25 \text{ mm}$, where the L_n measuring section was 4 mm . Cut-off filtration was used in the analysis of the surface roughness results.

After the friction tests the samples were once again used to produce four additional profiles for each sample. Three were done perpendicular to the friction groove and one profile was done along the groove at the point of the largest depth change. Having profilograms made across the groove, it was possible to determine the average groove depth for a given sample group, representing wear. Results were also

scattered by specifying half-intervals of confidence for the average value at the significance level of 0.1.

3. Results and discussion

3.1. Preliminary tests for friction components made of various materials

Preliminary tests were performed for the high viscosity lubricant taken directly from the engine oil sump (label S0). The main goal was to recognize the lubricating properties of the product for wide range of materials typically used as friction components. Aside from cast-iron we used aluminum, brass and polymer samples sliding on the steel ring of the Amsler tribometer (Fig. 2). In this trial the test time was reduced, down to 1 min for each sample, the speed and load were 200 min^{-1} and 1100 N respectively. For most sample materials the high viscosity lubricant S0 couldn't reach the lubricating performance of the high quality commercially available grease. As a result higher wear was measured on the samples, especially for aluminum (Fig. 2). This could be explained with low adhesion of the CNT based grease to the metal surface. Remarkably, better results were obtained for the tested polymer, a material widely used as friction component in technical application. Thus, in the following tests we focused on polymer friction components.

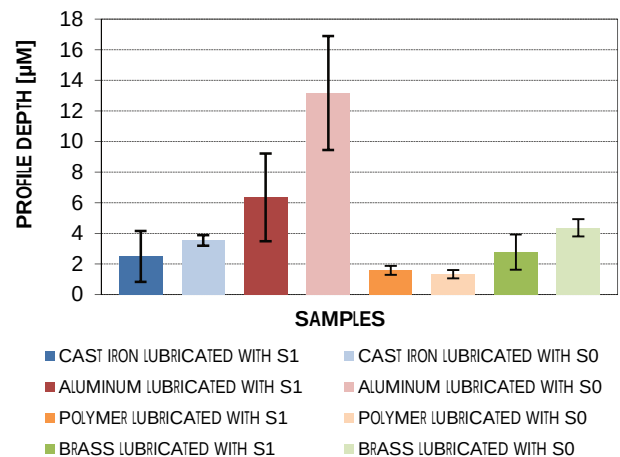


Fig. 2. Comparison of lubricants S0 vs. S1; wear measured after one minute friction tests on the surfaces of samples produced from different materials; mean values and confidence intervals for mean values at significance level of 0.1

3.2. 15-minute tests for polymer surface

Three samples and rings were prepared for each of the lubricants tested. The lubricant was introduced by covering the friction surface of the sample and the ring with a given lubricant achieving a layer with a thickness of at least 2 mm before the friction test began, and then additionally covering the friction surfaces of just the rings during the friction tests at 6 min and 10 min in. The friction test duration was set at 15 min. During the friction test, resistance to motion and temperature were measured using a thermocouple under the surface of the sample at a depth of about 3 mm . Measurements were made at 50 s, 2 min, 5 min, 7 min, 10 min, 13 min and 15 min. During the 15-minute test, the ring speed was $v = 228 \text{ rpm}$ and the load $F = 1200 \text{ N}$.

The friction test with the use of lubricant S1 had a clear impact on the reduction of the surface roughness, for the material of the tested plastic, which defined by parameter Ra reached an average value of $0.11 \pm 0.03 \mu\text{m}$ (Fig. 3). When analyzing the result of surface roughness measurements, the presence of S5 lubricant had a clear influence on the increase of the tested material surface roughness, as seen by the increased average value of the Ra parameter. This takes place both in relation to the friction test involving the lubrication of this material with S1 and in relation to the surface roughness of the tested plastic at the beginning of the friction test. The value of Ra after a 15-minute friction test in this case can on average reach $0.33 \pm 0.05 \mu\text{m}$.

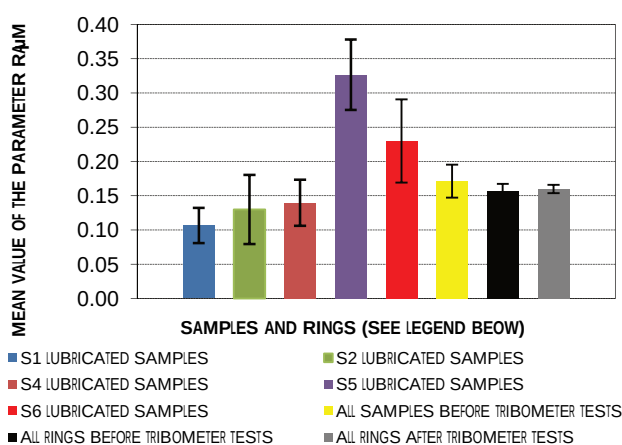


Fig. 3. Ra parameter for samples and rings before and after tests in the Amsler tribometer; mean values and confidence intervals for mean values at significance level of 0.1

The average Ra parameter values for rings before and after the friction process showed no differences. This may indicate the lack of impact of any of the tested lubricants on the wear of the steel spigot in the plain bearing when operating with the tested plastic in simulated friction conditions.

It should be noted that there are no differences in surface roughness (no difference in parameter Ra) of the friction surface of the samples and rings prior to the friction tests, which minimized the friction test preparation, which is beneficial from the point of view of the test results.

The use of selected lubricants S2, S4 and S6 in the tested friction setup (in conditions determined by the experiment) did not affect the increase or decrease of surface roughness after the tests.

The results of the average Ra parameter values along with the result data spread also indicate no difference between the roughness on the friction surface of the tested material with the use of lubricants S5 and S6. Figure 4 shows the view of one of the samples lubricated with S6 grease. The friction surface indicates the presence of parallel scratches that may indicate abrasive wear (Fig. 4b).

The results of average groove depth value measurements of the surface roughness profiles (Fig. 5) being a measure of wear, indicate no differences in the wear level of surfaces lubricated using S2, S4 and S6. It should be emphasized, however, that the average values of wear indicate a greater wear of the tested material in the friction tests

when using the previously mentioned three lubricants than in the case of S1.

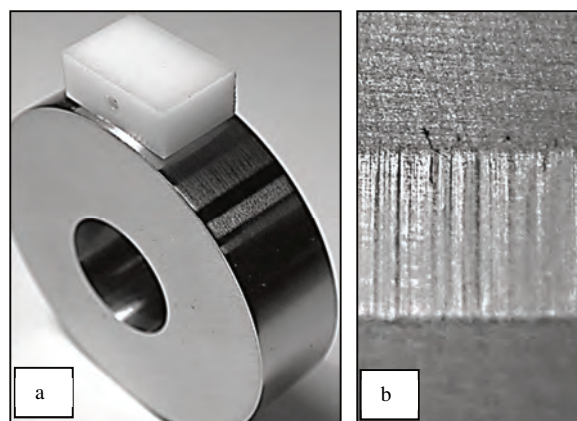


Fig. 4. Amsler tribometer: friction components (a) and scratches on the polymer sample after S6 lubricant test (b)

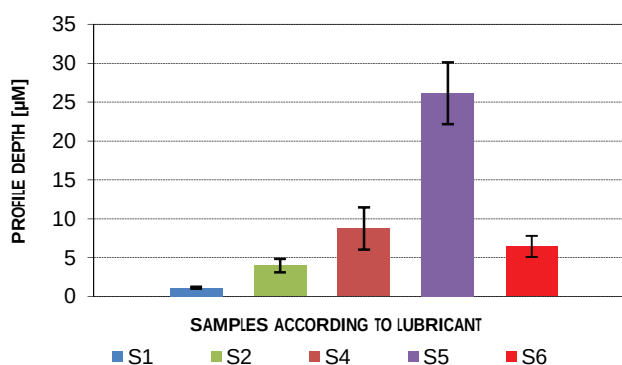


Fig. 5. Wear measured on the polymer samples after test performed for various lubricants; depth of the friction track profile; values and confidence intervals for mean values at significance level of 0.1

The wear results unambiguously indicate that the smallest wear in the simulated contact during sliding friction was observed when using lubrication with S1.

The use of S5 in the tests resulted in the highest values of sample wear in relation to the other lubricants tested (Fig. 5). The unfavorable lubricating effect of the S5 lubricant is further confirmed by the significant sample temperature increase as a result of the friction, Fig. 6.

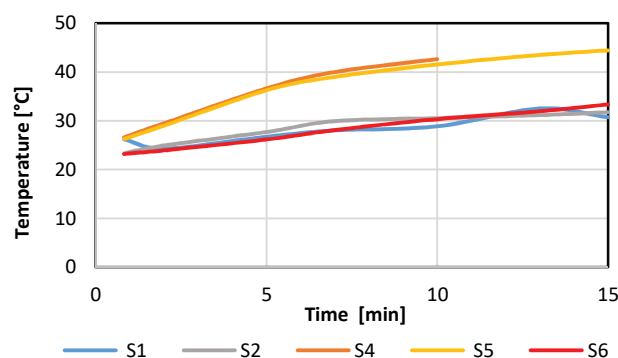


Fig. 6. Temperature in the sample material during friction test in tribometer; mean values for tested lubricants

3.3. 30 minute tests for polymer surface

For the polymer samples an additional test was performed to compare the best CNT based lubricant to the standard commercial one, with results shown in Fig. 7.

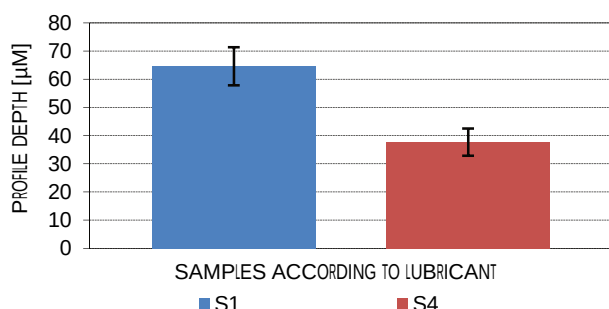


Fig. 7. Mean wear values measured as groove profile depth; comparison for S1 and S4 lubricant tested on polymer samples; mean values and confidence intervals for mean values at significance level of 0.1

Similarly to the 15-minutes test the speed of the ring was $v = 228$ rpm, the load in Amsler tribometer was set to $F = 1200$ N, but the operating time was extended to 30 minutes. The S1 lubricant that previously provided the best results failed in this prolonged test, leading to an extremely high wear characterized by groove depth of over 64 microns. For the same friction conditions the CNT based lub-

ricant lead to a groove no deeper than 38 microns. We hypothesize a CNT related mechanism of anti-wear protection which still needs to be explained.

4. Summary

The presented preliminary tests of high viscosity lubricant formulated by commercially available engine lubricating oil with the addition of CNTs allowed us to reveal a unique, strong interaction of CNTs and the polymer surface. For long-duration friction tests the wear observed on polymer sample surface lubricated by CNT based grease was nearly 50% smaller than for the reference high quality lithium complex grease. On the other hand, friction tests performed for the CNT based lubricant on metal samples failed due to low lubricant adhesion to metal surfaces.

For all tested surfaces we observed a strong impact of the CNT based lubricant formulation on the wear results, this justifies further experiments targeting the role of nanocarbon morphology in friction processes. In general the obtained results are necessary for better understanding of the CNTs role in the friction processes. Therefore extended studies of the CNTs based lubricants rheology and microscopic structure as well as a characterization of the wear grooves are necessary.

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Analysis of the availability of aircrafts with alternative propulsions

Global air traffic doubles every 15 years. Therefore, aviation emission grows as well. It is advisable to look for an alternative aircrafts' propulsion for both commercial and general aviation (usually flying closer to the ground). There are many prototypes proposed by homemade projects, through companies wanting to extend their offer. There are also several more advanced constructions ready to self-set or serially produced. In the article the main topic are electric aircrafts as an example of possibility to reduce the toxic compounds emission. According to the analysis some of companies specializing in unmanned aircraft production, starting to expand their business with manned products. In the article, analyses of electric aircrafts different models, which are available on market and presented the comparisons characteristics based on e.g. cruise speed, range for every model or charging time were made and conclusions about alternative aircraft propulsions, were made.

Key words: aircrafts, alternative propulsions, electric aircraft

1. Introduction

For a long time, due to economic and ecological aspects, solutions have been sought that would minimize the use of conventional fuels in transport. Such solutions are used in various branches of transport, including road, rail and air transport [1]. The aircraft emission can cause deterioration in local air quality (near airports) and climate changes connected with gases trapping heat in the atmosphere and with formatting aerosols which lead to changes in clouds (at cruising altitude). Under the agreement of 192 members of ICAO, the global aviation emission target is 50% reduction by 2050, however without state-of-the-art solutions it will be hard to achieve. Over the past decade, more than 70 all-electric conceptual, experimental and commercial aircrafts have been researched, with a particular focus on light aircraft [2]. Currently, in 2019 there are several models already available for production. The highest interest in electric aircrafts in Europe is in Norway. In this country, it is planned, that all short-haul flights (up to 1.5 hours) would be carried out by electric aircrafts by 2040. The first state company, to resign from aircrafts with combustion engines propulsion is Norwegian Avinor. The first flight tests of a small 19 seats passenger airplane are to begin in 2025 [3]. Furthermore, the EasyJet airline, the third world biggest low-cost carrier, announced a cooperation with American Wright Electric company in 2017. The goal of the collaboration is to create a full electric commercial airliner. Initial flights are scheduled for 2027 [4].

Nowadays, no electric commercial airliners are produced, which is way the characteristic will be based on ultralights, motor gliders and motor hang gliders. The main producer of ultralight electric aircrafts is Pipistrel. It is a Slovenian company founded in 1989 with headquarters in Ajdovscina [5]. Pipistrel is a producer of two ultralight aircraft with electric propulsion: Alpha Electro, produced since 2015 and Taurus Electro produced until 2011. The next company on the electric aircraft market is Electric Aircraft Corporation. It is an American aircraft manufacturer that produces under the brand name Electraflyer. The company is currently producing two model of aircraft with electric propulsion: the motor hang glider (Electraflyer

Trike, which has been in production since 2008 and the moto glider (ElectroFlyer ULS), which has been produced since 2012 [6]. The Alisport company also manufactures the motor gliders. It is an Italian manufacturer that deals with the creation of aircraft ready for flight and sets for amateur production. The model produced by the company is Silent 2, which is powered by solar panels [7].

2. Characteristics of aircraft models with electric propulsions

The first electric drive airplane to be described is Alpha Electro (Fig. 1). It is a training aircraft belonging to the group of ultralight aircraft. Alpha Electro has a wingspan of about 10.5 m, has a two-seater, T-tail, high wing motor plane. This model is made almost entirely of composite materials. The aircraft is equipped with a solid three-wheeled chassis, including wheels with breaks, a U-shaped composite rail and controllable wheel [5, 8]. Alpha Electro features flaperons, one movable surface on each wing, which simultaneously a flap and an aileron. Full dual main flight control levers make the Alpha Electro ideal for initial and advanced flight training. All aileron, elevator and flap controls are connected to the cabin controls using self-fitting push-pull tubes (Fig. 2). The rudder is controlled via cables. The elevator trim is a mechanical, spring type mechanism. H-type safety belts attached to the fuselage at three mounting points are the risk minimizing features.



Fig. 1. Ultralight aircraft Alpha Electro [5]

The rudder pedals can be adjusted before and also in-flight to suit everybody's dimensions and ergonomic requirements. The aircraft is equipped with two battery boxes. One is situated behind the cabin bulkhead and the second one is behind the firewall. The battery system is ventilated in order to prevent overheating. The powering unit is a 60 kW electric motor capable of energy recuperation during the descent. The doors, windshield and top window are made of 2 mm anti-UV tinted Lexan, which was specially designed not to shatter or split on impact. The main wheel disk type, hydraulic brakes are activated via a cockpit hand-lever. The cabin is ventilated through the special vents fitted into the doors. The propeller is a fixed pitch three-blade design.

Due to safety reasons, the resettable electric circuit brakes enable testing individual circuit items and disconnect the batteries if needed. The full set of lights is mounted: navigational lights (NAV), anti-collision lights (AC) and a landing light (LDG).

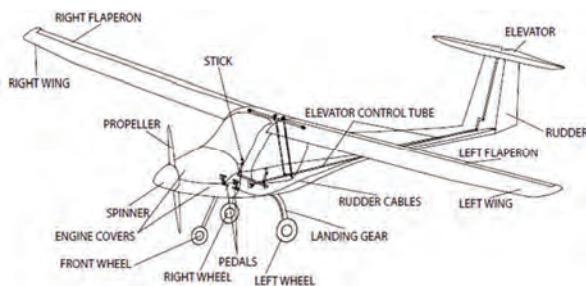


Fig. 2. Alpha Electro construction schema [7]

The next model to describe is a plane from the same manufacturer – TaurusElectro (Fig. 3). This model has got a 15-meter wingspan, T-tail and is made almost entirely of composite materials. The wing configuration is a mid-wing, the propulsion system is fully retractable in order to increase the sliding performance. The aircraft is equipped with a taildragger fully extended gear consisting of two main wheels with brakes [9]. The tail wheel is controlled by the pilot in cockpit. Like the Alpha Electro, Taurus is equipped with flappers, which are combination of flaps and ailerons, located in one deflecting surface. The flaps have 3 setting options: neutral, start and landing. All ailerons, lift and flaps are connected to the cab to be able to control using self-adjusting push-pull tubes. The lift has a mechanical trimmer placed inside the fuselage to prevent too much resistance [9].



Fig. 3. Taurus Electro motor-glider [10]

All the models are equipped with H-type seat belts mounted to the fuselage in three points. The batteries are placed inside the fuselage in four metal boxes with dedicated connectors. The charging process is available only by original chargers by Pipistrel. The cabin of TaurusElectro is transparent or color in blue (Fig. 4). The main brakes are hydraulic [9].



Fig. 4. Taurus Electro motor-glider model [9]

The cabin ventilation is carried out through special cables mounted on the cabin frame and can be adapted depending on the needs of the crew. To improve the aerodynamics of gliding flight, Taurus Electro has a fully automated engine start procedure. It is carried out using the switch on the dashboard. The basic instruments are installed with predetermined operational limits. The Taurus can be equipped with a ballistic parachute rescue system installed optionally (in some countries, e.g. Germany, ballistic rescue systems is required).

The next characterized model is ElectraFlyer Trike (Fig. 5). It is an ultra-light motor hang glider, equipped with an electric motor instead of a traditional gasoline engine, created by an American manufacturer – Electric Aircraft Corporation [11]. The ElectraFlyer Trike is built in accordance with the Federal Aviation Regulations, part 103. The standard ElectraFlyer is equipped with a 5.6 kWh polymer lithium-ion battery that powers a 19 HP (14 kW) electric motor.



Fig. 5. Motor hang glider ElectraFlyer Trike [6]

One of the possible modernization options is changing the engine for 40 km (30 kW) power. Charging time from standard (in USA) plug-in (120 V) depends on the number of batteries packs and ranges between 5 and 6 hours. Charging time can be shortcut to 2 hours with 240 V socket. With Stratus wing model, aircraft gross weight can be reduced to 113 kg. There is a possibility of installing an emergency parachute.

The next electric aircraft to be described is ElectraFlyer – ULS (Fig. 6), manufactured by the same company. ElectraFlyer – ULS is projected in accordance with FAR rules, part 103 (Ultralight Vehicles) [12]. In pursuance of the empty weight limitation – category 115 kg – of these rules, the motor-glider's empty weight is 111 kg. It's a mid-wing construction, one-seater, stall three wheeled gear and single electric engine [12].



Fig. 6. Electra Flyer – ULS [13]

The glider is made of composite materials – carbon fiber and foam. The engine and the propulsion system are supplied by electric aircraft corporation, while the airframe is built in cooperation with Airsport in Czech Republic and is adapted from the existing project – Airsport Song. The propulsion system includes a Fishman 20 HP (15 kW) engine, an electronic controller and a 3.3 kWh Li-Po battery pack [12]. Optionally, the model can be equipped with an additional battery with a capacity of 3.3 kWh, which gives two hours of durability. A two-blade propeller with a fixed pitch is standard equipment. Additionally, the motor glider can be equipped with a folding propeller made of carbon fiber.

The next described aircraft is Silent 2 Electro. It is a single-seat, self-launching microlight sailplane. The aircraft is made of carbon fiber and glass-fiber, has a wingspan of 13.5 m (Fig. 7) and "T" tail configuration [7].

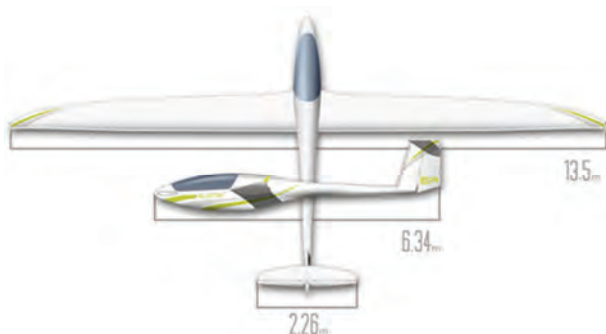


Fig. 7. Motor-glider Alisport Silent 2 dimensions [7]

The control system consists of a speed brake, a flap-ers, rudder, variable incidence tail plane and elevator are operated by a system of pulleys, cables and pushers. The glider's tail is connected, via a cable, to the flap control. The pilot's seat, which is not adjustable, is covered with a two-piece pillow connected by a zip fastener, which is attached to the seat base and at the back by Velcro fasteners. The glider is equipped with adjustable rudder pedals.



Fig. 8. Alisport Silent 2 [13]

The canopy is attached to the glider by a hinge at the front edge of the canopy frame (Fig. 8). It is locked by means of two locking pins on each side of the rear part of the frame. The left locking pin supports a microswitch mounted on the inside of the fuselage that provides the "canopy closed" signal to the controller. The front Electric Self-launch system consists of two lithium-polymer batteries, a controller, a control unit, a 22 kW brushless motor and two fixed pitch propellers. The electric motor is located in the nose of the aircraft and when it does not work, the propeller blades, are folded back. Two lithium-polymer batteries are connected in series and are placed in a compartment behind the cockpit. When fully charged, the batteries have a total capacity of 4.3 kWh. Propeller rotations are set using the throttle/brake. The distance necessary to take off on the asphalt is 140 m, and the engine provides a climb speed of 2.0 m/s (400 ft/min) [7].

The last discussed model of an electrically powered airplane is MC E15 Cri-Cri (Fig. 9). It was designed by a French engineer Michel Colomban. The production, the model deals with the European Aeronautic Defense and Space Company. It is the smallest single-seat ultralight aircraft.



Fig. 9. MC-15 Cri-Cri airplane [14]

It is made of metal construction and light composite materials. It has a T-tail. The Cri-Cri was the world's first electric-powered aircraft.

The aircraft's propulsion consists of four brushless 2JPX PUL 212 electric motors, each with 15 HP. The energy is provided by lithium batteries with a capacity of around 4 kW.

Wingspan of this model is 4.90 m, length 3.90 m and height 3.10 m (Fig. 10). The weight of the aircraft is 80 kg, and the maximum weight is 170 kg. Cri-Cri can climb at a maximum speed of 3 m/s and develop speeds up to 220 km/h [16]. The aircraft can perform a flight for about 30 minutes.

Apart from described models, there are also other manufacturers constructing electric aircrafts e. g.: Solar Impulse, Magnus or Rutan [17]. However, these are prototypes, university projects, aircrafts on the development stage, or special purpose constructions, thus they are not a part of this comparison.

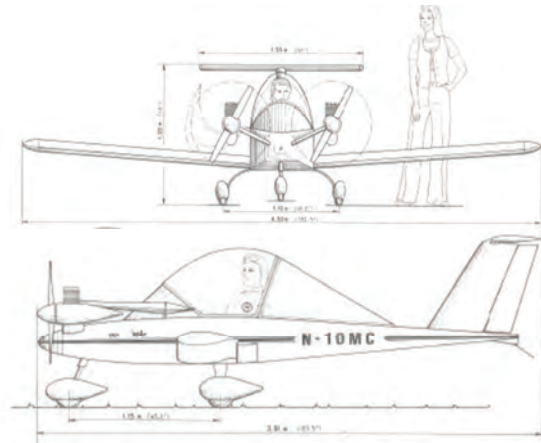


Fig. 10. MC-15 Cri-Cri airplane dimensions [15]

3. Comparison of the solutions available on the electric aircrafts' market

In order to perform the analysis of available electric aircrafts solutions, the assessment criteria were chosen. The criteria available for almost every aircraft are: number of engines, engines power, batteries capacity, cruise speed, flight time, Maximum Take Off Weight (MTOW) and charging time for two different voltage chargers (Table 1).

Table 1. Comparison of aircrafts models according to parameters available

Aircraft model	Aircraft type	Engine count	Engines power [kW]	Batteries capacity [kWh]	Cruise speed [km/h]
Electraflyer Trike	hang glider	1	13.5	5.6	48.4
Electraflyer ULS	ultralight motor glider	2	27	6.6	64.4
Pipistrel Alpha Electro	airplane	2	50	42	157
Pipistrel Taurus Electro G2	motor glider	4	54	22.8	163
Silent 2 Electro	motor glider	1	13.5	4.4	70
MC15E Cri-Cri	airplane	2	27	4	110

Aircraft model	Aircraft type	Flight time [h]	MTOW [kg]	Charging time – 110 V charger [h]	Charging time – 220 V charger [h]
Electraflyer Trike	hang glider	1.5	112	6	2
Electraflyer ULS	ultralight motor glider	2	238.14	4	1
Pipistrel Alpha Electro	airplane	1.25	550	8	1.6
Pipistrel Taurus Electro G2	motor glider	2	450	5.5	3.5
Silent 2 Electro	motor glider	1	300	–	–
MC15E Cri - Cri	airplane	0.42	170	–	–

In case of two models: Silent 2 Electro and MC15 Cri-Cri, it was not possible to find information on the charging time. All the models discussed are in production, but very often (e.g. Electraflyer Trike and ULS) are manufactured directly for the customer, according to his order. Thus, there are very few of them in use. Electraflyer models are often delivered in separate parts, so that the customer could connect them himself, which means these are models for people interested in broadly understood avionics and aircraft construction.

Next, in order to make an analysis, the characteristics based on the data contained in table 1 will be discussed.

Figure 11 shows the take-off mass for individual aircraft model. The take-off mass is stated in kilograms, varies from 112 kg to 550 kg and depends mainly on the size of the model and its construction.

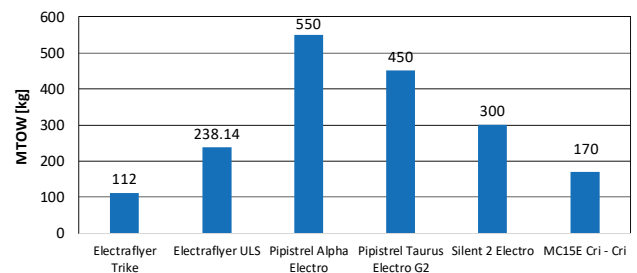


Fig. 11. The Maximum Take Off Weight bar chart

The Pipistrel Alpha Electro has the largest MTOW from all the aircrafts mentioned (Fig. 11). It's 550 kg. Its take-off mass is justified by its size – it is the largest of the models discussed and belongs to ultralights which distinguishes it from other moto-hang gliders or motor gliders. The second model with a big take-off weight is the motor glider, by the same manufacturer – Taurus Electro G2. It has a take-off weight of 450 kg. Silent 2 Electro, also a glider, created by the Alisport manufacturer, is the next aircraft with a slightly lower take-off weight. It is slightly smaller than the Pipistrel motor glider – the difference in the wingspan is about 1.5 meters. The next motif is Electraflyer ULS, whose MTOW is 283 kg, while the weight of an empty aircraft is 111 kg. The MC-15E Cri-Cri model has a take-off weight of 170 kg. The plane itself weighs 80 kg, and is the smallest of the aircraft in question with a wingspan of about 5 meters. The lightest among the discussed ones, is the ultralight Electraflyer Trike. The take-off weight of the hang glider is less than 113 kg.

The following is a description of the engine performance, number of engines and the speed of the aircraft models in question (Fig. 12). The power shown refers to the sum of the power generated by all the engine, i.e. if the Electraflyer ULS generates 27 kW and has two motors, then each motor produces 13.5 kW.

The Pipistrel Taurus Electro G2 has got the biggest power coming from 4 engines, which all together gives 54 kW power (Fig. 12). Furthermore, the ultra light plane – Alpha Electro produces about 50 kW of power, using 2 engines, or about 25 kW each. The Electraflyer ULS motor glider and the MC15E Cri-Cri ultra-light aircraft generate about 27 kW of power using two powering units. The Motorminer Silent 2 Electro and the Electraflyer Trike are equipped with one electric motor that provides 13.5 kW of power. As can be seen, most of the aircraft models in question have 13.5 kW motors, only in different quantities. Only in the case of the ultra-light Alpha Electro aircraft, two 25 kW drive units were used. It is known that, the speed of any means of transport with an electric or combustion engine depends mainly on the engine power. An important parameter in terms of the speed achieved is also the takeoff weight of a given type of aircraft. The aircraft with the highest cruising speed is moto glider Taurus Electro G2, whose performance is 163 km/h.

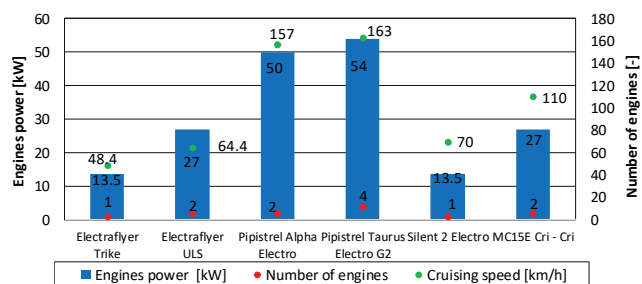


Fig. 12. The number of engines and their power

The ultra-light Alpha Electro model and the MC15E Cri-Cri are also characterized by a great value of cruising speed compared to other models. These aircraft reach 157 km/h and 110 km/h respectively. The aircraft with the lowest cruise speed is Electraflyer Trike. However, the type of this aircraft should be taken into account. It is a motor hang-glider, so achieving high cruising speeds is not a priority. Two aircraft models: the MC15E Cri-Cri and the Electraflyer ULS have the same engine power, but achieve different maximum cruising speeds. It is caused by the mass of these two models. The Electraflyer motor glider is 70 kg heavier than the ultra-light MC15E Cri-Cri.

Figure 13 presents the dependence covering three series of data: battery capacity for the discussed aircraft models, flight time and power of installed engines. The flight time depends primarily on the battery capacity and the power generated by the electric motors. If the capacity of the batteries is too small in relation to the power that the engines produce, the flight range will not be long. This is the case for the MC15E Cri-Cri model, which is able to stay in the air for only about 30 minutes, because the capacity of its batteries does not last long enough to generate installation power in the aircraft's propulsion units.

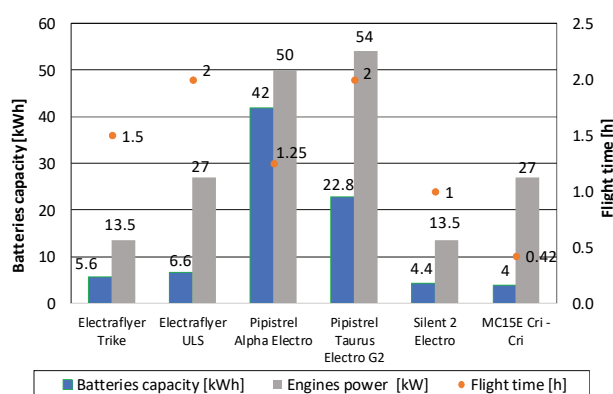


Fig. 13. The flight time, batteries capacity and engines power chart

The longest flight time is 2 hours and is possible to achieve flying Pipistrel Taurus Electro G2 and Electraflyer ULS as well (Fig. 13). Both models differ significantly in the capacity of batteries and engine power – Taurus Electro has about 3.5 times more battery capacity and twice as much engine power. As it can be noticed, the Electraflyer ULS can stay in the air as long as the Taurus Electro G2, despite the much smaller capacity of the batteries. This is due to the power of engines, Electraflyer ULS will have much lower performance associated with, for example,

cruising speed compared to the Taurus Electro G2. Aircraft that also provides a long flight time is the Electraflyer Trike. When charging to 100%, air operations can be performed for about 1.5 hours.

Figure 14 shows the cost of charging these aircrafts. The prices were calculated only for four of the six models discussed, because there are no information on charging models MCE15 Cri-Cri and Silent Electro 2. The Pipistrel company has its own charging station dedicated for two discussed aircraft models – ultralight Alpha Electro and a Taurus Electro G2 motor glider. The price for charging in a dedicated station is 2 € for 1 kWh. Referring the price to the battery capacity, the results presented in the chart were obtained. The charging costs for Electraflyer Trike and Electraflyer ULS were calculated using data from the manufacturer – price 0.53 € per 1 kWh. The largest costs of using are in case of Alpha Electro. This is due to the large capacity of the battery, which is enough to charge 2 electric motors that keep the plane in the air. The lowest cost of charging is characteristic for the Electraflyer aircraft models. Both models are equipped with low-capacity batteries and the charging rate is very small compared to the price of the Pipistrel charging station. In addition, the charging costs of the models in question were converted into Polish zlotys (PLN). The calculations were made in accordance with the Euro rate as at 14/02/2019. The calculations related to the exchange rate were made to better present the costs for possible Polish users and, to summarize the work, be able to compare the prices of aircraft usage with those applicable to car vehicle users.

The operating costs are significantly different from those connected with piston engines aircrafts usage. For example, the basic airplane Private Pilot License (PPL(A)) training lasts at least 200 hours. A very popular training airplane (Cessna 172 P) consumes about 40 l of fuel. The price for 1 liter of AvGas is about 8.12 PLN it gives 64,960 PLN for fuel per training PPL pilot. If the pilot would be trained at Alpha Electro the price for training would be 58,195 PLN. What gives more than 6,500 PLN of savings. It can be stated that flying an electric aircraft is not only advisable for ecological reasons, but also economical ones, which is usually more appealing argument.

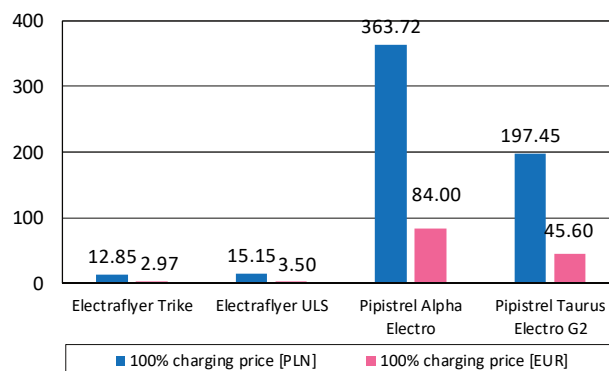


Fig. 14. Full charge costs

4. Conclusions

Electric aircrafts can deliver an opportunity for high-speed journeys between city centers in the future [18].

In the article, the analysis of available aircrafts with electric propulsions were made. It can be stated, that in the last few years there has been a significant development of work carried out in this field. Currently, several models of fully electric aircraft are available. These are mainly ultralight aircraft and powered gliders. There are many works related to the production of electric aircraft, but none of the prototypes is officially produced. These models are now used not only for recreational aviation but also for training. Some of the manufacturers also have special charging stations for ultralight aircraft. In addition, the prices charged for one full charge of an ultralight aircraft are comparable to the costs of charging a car with an electric SUV with a high capacity of batteries. Which means, they can sooner, or later replace cars on interurban travels. The only disadvantage of aircraft with electric drive is a limited flight time, which in case of using batteries and electric motors is very short and allows to perform air operations in the air for a maximum of 2 hours.

The biggest advantage of all the models of transport modes discussed is the zero or significantly reduced carbon dioxide emission and the very low level of noise emitted. Nowadays, actions for sustainable development are a very important aspect. The society should be aware of the importance of reducing pollutants emitted to the atmosphere by human beings and through actions related to the design and development of transport means with alternative propulsion sources give an example and inspiration to care for the common environment without negatively affecting the development of future generations.

Nomenclature

AC	Anti-Collision lights
ICAO	International Civil Aviation Organization
LND	Landing light
EU	European Union
FAR	Federal Aviation Regulation
GPS	Global Positioning System
HP	Horse Power

MTOW	Maximum Take Off Weight
NAV	Navigational lights
PPL(A)	Private Pilot Licence (Airplane)
ULS	Ultra Light Series
USA	United States of America
UV	Ultra-Violet

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Influence of cetane-detergent additives into diesel fuel with RME content increased to 10% on the parameters of indicator diagrams and rate of heat release in a diesel engine

This publication is the next part of the article "The influence of cetane-detergent additives in diesel fuel increased to 10% of RME content on energy parameters and exhaust gas composition of a diesel engine". The cause-effect analysis of the phenomena related to the impact of 3 additive packages used in diesel oil with RME content increased to 10% (compare to standard diesel fuel with 7% of RME) was described. The basis for the analysis of the impact of the tested fuels on energy parameters and composition of exhaust gases were the parameters of indicator diagrams and heat release parameters. It was found that the first set of additives affects the delay of auto-ignition of fuel and kinetic fuel combustion speed only at low engine loads. In this range of engine operation the NO_x concentration in the exhaust gas is low and besides there is a large of EGR. The second additive package was operated at high engine loads but its impact on the lower self-ignition delay was quantitatively small. Therefore, in the third packet of additives, the amount of additives used in the second packet was doubled. Then a satisfactory shortening of the self-ignition delay and reduction of the max rate of kinematic heat release was achieved as a reason of a reduction of NO_x concentration in the exhaust up to 8% (compared to the reference fuel).

Key words: cetane-detergent additives, RME, indicator diagram, heat release, diesel engine

1. Introduction

The paper is the second part of the article *The influence of cetane-detergent additives in diesel fuel increased to 10% of RME content on energy parameters and exhaust gas composition of a diesel engine* and is a cause-effect analysis of the phenomena related to the impact of 3 additive packages used in diesel oil with RME content increased to 10%.

In the studies described in the first article, it was found that the first fuel additive package reduces the NO_x concentration in a beneficial, though the insignificant way in the exhaust gases only at low engine loads. The second additive package quantitatively influenced the reduction of the NO_x concentration in the engine exhaust in a similar way as the first additive. However, the effect of this additive was the more effective (from the point of view of reducing the NO_x concentration) the higher the engine load. Due to the fact that the second additive package has been practically too small, it was decided to increase the amount of cetane additive in package 3 by a factor of 2.

The article analyzes the correlation between the number and proportions of additives used in packages 1, 2 and 3, and the parameters of indicator diagrams, as well as parameters of heat release graphs (including the maximum rate of kinetic and diffusion heat release), NO_x concentration and smoke opacity. This operation allowed both the determination of the right direction of quantitative and qualitative changes in relation to the value of engine energy parameters and the composition of the exhaust for the engine fuelled by tested additive packages.

2. Characteristics of tested fuels and research methodology

Three Innocet[®] detergent-dispersing packages (A, B, and C) were used. The B10 diesel fuel (10% (by vol.) of RME), whose physico-chemical characteristics are shown in Table 1, was enhanced with the A, B, and C Innocet[®] packages in the quantity of 1,500 mg/kg. Such enhanced

B10 diesel fuel had cetane number equal to 58. The Innocet[®] A, B, and C packages contained 50 mg/kg of a commercial detergent-dispersing additive and differed by the level of dosing of the stabilizers taking synergistic part in creation of free radicals in the area of low-temperature and indirect oxidation of the B10 diesel fuel. In this manner, 4 fuels were created, which then underwent engine tests:

- TM/134/18 – comparative fuel B7 (7% (V/V) RME)
- TM/135/18 – B10 fuel with the Innocet[®] A package
- TM/136/18 – B10 fuel with the Innocet[®] B package
- TM/154/18 – B10 fuel with the Innocet[®] C package

The engine tests of the fuels in question were performed at the Internal-Combustion Engines Laboratory of the Institute of Motor Vehicles and Internal-Combustion Engines of the Cracow University of Technology.

Tests for each of the tested fuels were carried out using a 4-cylinder diesel engine, type 1.9 TDI (AJM version), made by VW, whose technical characteristics are shown in Table 2.

In order to measure and analyze the parameters of the engine fuelled sequentially by tested fuels was used part of the measuring systems, which are equipped laboratory of Cracow University of Technology. The study was carried out using the popular VW 1.9 TDI engine in the AJM version (with unit injectors), popular in Europe, in passenger cars and light commercial vehicles. This article describes the influence of engine load (at max torque speed 2000 rpm) on the basic engine parameters and exhaust composition. In the study were used Schenk engine dynamometer, AVL Fuel Balance (for measure of fuel consumption) and AVL Bench Emission System CEB II for measuring the concentration of gaseous components. In addition, the air temperature collected by the engine, the air temperature behind the turbocharger, behind the inter-cooler and into the intake manifold (behind the EGR valve) were measured. The exhaust gas temperature was measured before and after

the turbocharger. For comprehensive analysis, the impact of the engine load on the boost pressure and the start of fuel injection were measured. In the second part of the article, the AVL Indimeter 617D was also used to determine and analyze the fast-changing pressure in the engine cylinder (as a function of the engine crankshaft rotation angle) and to calculate the rate of heat release.

3. Delay of self-ignition

The process of delay of chemical self-ignition continues to be the object of fundamental research consisting in modelling of the kinetics of advanced chemical reactions of oxidation of hydrocarbons. The process of oxidation of hydrocarbon fuels covers hundreds of grades of hydrocarbon compounds participating in thousands of chemical reactions which makes the entire process of low-temperature, transitional, and high-temperature oxidation very complex.

The chemistry and the delay time of self-ignition depend on the molecular structure of the diesel fuel and on the kinetics of the elementary reactions of low-temperature oxidation of hydrocarbons.

The process of self-ignition of diesel fuel is initiated by a start of reactions that generate free radicals produced from stable hydrocarbon compounds. The progress of branched chain reactions increases the number of free radicals involved in the self-ignition.

The chain reaction theory was developed and published by Nikolay Semyonov for which, together with Cyril Hinshelwood, he won the Noble Prize in Chemistry in 1956 [2].

Curran et al. presented [3] ten main categories of reactions involved in the process of chemical self-ignition:

- I. Unimolecular decomposition (molecular elimination reaction);
- II. Hydrogen atom transfer;
- III. Oxygen addition reactions;
- IV. Other addition of H^* , $^*OH$, HOO^* radicals and radical species to olefins;
- V. Recombination reactions;
- VI. β -decomposition;
- VII. Isomerization reactions;
- VIII. Creation of double bonds by elimination of two substituents in a one-stage reaction mechanism;
- IX. Free-radical cyclization;
- X. Reaction of disproportioning of two radicals.

The mechanism of low-temperature oxidation of hydrocarbons and of the self-ignition chemistry is also presented in the publication [4].

Self-ignition of diesel fuel takes place when the energy released by free-radical chain reactions results in an exponential increase of the temperature of the load in the engine cylinder, which of course is associated with a pressure surge in the cylinder. In the presented laboratory engine tests, this is how the start of self-ignition of the fuel was determined. Self-ignition delay, t_s , is defined in the publication as the angle of rotation of the engine crankshaft (determined with the accuracy of 0.1 deg) between the start of fuel injection (at a constant rotational speed and load on the engine – the same for all fuels) and the start of self-ignition of the fuel (determined based on the changes of the $dp/d\alpha$ in the engine cylinder). Such self-ignition delay (t_s) is only

a part of low-temperature oxidation of hydrocarbons and of the chemical part of self-ignition; however, the error made in the assumption is the same for all the tested fuels. This enables an analysis of cause and effect phenomena associated with use of the reference fuel and the fuels containing cetane-detergent additives.

4. Experimental

4.1. Parameters of indicator diagrams

Because the impact of engine load on most of the analyzed parameters is much higher than the impact of the tested fuel additives, a linear load characteristics analysis of the parameters of the indicator diagrams and the heat release speed is very difficult. Therefore, in order to perform a more accurate analysis, bar graphs were prepared, based on cross-sections of load characteristics, of all parameters for low, medium, and high load of the engine (50 Nm, 150 Nm, and 250 Nm). As a result, at changed value ranges (different for different engine loads) of specific parameters, it was easier to analyze the impact of the tested fuels on the analyzed parameters.

Historically, the oldest method of cause and effect analysis of fast-changing processes inside the cylinder of a combustion engine is its indication. Measurement of the pressure of the working medium in the engine cylinder as a function of the angle of rotation of the crankshaft continues to be used in order to analyze the processes taking place in the cylinder.

The measured indicator diagrams for the four fuels that were used and the seven engine loads (at a constant rotational speed that is characteristic for the maximum torque of the engine $n = 2000$ rpm) were used to determine a number of parameters that facilitated an analysis of the impact of fuel additives on the energy parameters and the composition of the exhaust gas. Those parameters include:

- start of fuel injection α_{pw} ,
- start of fuel self-ignition α_{psp} ,
- fuel self-ignition delay τ_{sp} ,
- maximum rate of increase of combustion pressure $(dp/d\alpha)_{max}$,
- maximum combustion pressure $p_{s,max}$,
- angle at which the combustion pressure is the highest $\alpha_{ps,max}$.

Moreover, the values of the pressure in the cylinder as a function of the crankshaft rotation angle (for the four fuels) were used to calculate the heat release speed in the engine cylinder, which is described in the next subchapter.

In order to determine the start of fuel injection (α_{pw}), a VCDS computer diagnoscope, intended only for controllers used in VW engines, was used. A device of this type identifies the start of fuel injection (α_{pw}) as a sudden increase of the value of the current controlling the opening of the fuel injector pumps. This is not the real start of fuel injection, because the needle of the injector pump is lifted 150–200 μs after the start of the increase of the current impulse. However, even the start of the lifting of the injector (injector pump) needle does not necessarily take place exactly at the same time as the start of fuel injection. One can imagine a situation where the actual injection starts (the fuel flows through the outlet openings in the nozzle) when

10% of the maximum lift of the needle for diesel oil is achieved. However, for example in the case of a raw plant oil, whose viscosity is 18 times higher than the viscosity of diesel fuel, at the same lift of the injector needle there will be no fuel injection. Moreover, the real start of fuel injection may or may not take place (at the same lift of the injector needle) depending on the difference between the fuel pressure and the back pressure (the pressure in the engine cylinder). Having the above phenomena in mind, in the tests it was assumed that the fuel injection start (α_{pw}) is equivalent to the start of the increase of the current control signal that opens the injector pump (the same viscosity of the tested fuels – the same error of determination of the α_{pw} for all the fuels). As shown in Fig. 1, the start of fuel injection (α_{pw}) determined in the same way for all the tested fuels is the same (within the measurement accuracy limits).

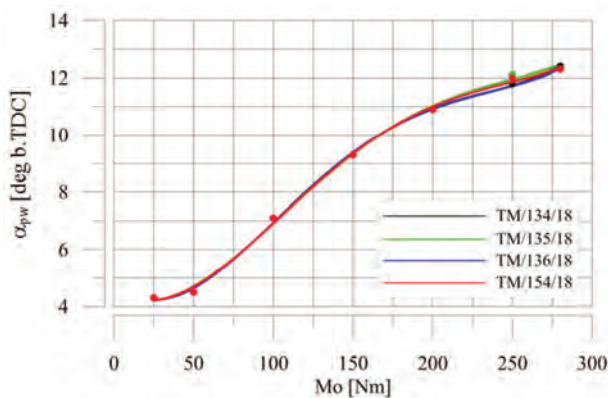


Fig. 1. Impact of the engine load on the start of fuel injection (α_{pw}) for the 4 tested fuels

Of course, as the engine load increases, the optimum fuel injection start is earlier, which is ensured by the engine controller, in the same way for all the fuels.

As mentioned in chapter 3, the start of fuel self-ignition (α_{psp}) in the engine cylinder was determined as the moment (angle of rotation of the crankshaft) when there is a pressure surge in the engine cylinder ($dp/d\alpha$). Similarly, as in the adopted method of determination of α_{pw} , also determination of the start of fuel self-ignition (α_{psp}) based on the indicator diagram involves a certain simplification because it does not take into account all phenomena and reactions taking place in the process of actual self-ignition of the fuel.

Literature on this topic contains examples of determination of the start of self-ignition of fuel in the engine cylinder based on the heat release speed (the first zero value of $\delta Q/d\alpha$). However, one must keep in mind that the values of $\delta Q/d\alpha$ are calculated based on very different computational models. Therefore, the start of self-ignition determined using this method, depending on the thermodynamic model, can differ by as much as 15%. Consequently, in this publication the start of fuel self-ignition (α_{psp}) was determined based on the changes of $dp/d\alpha$ as a function of the angle of the crankshaft, which for the tested fuels and the 3 engine loads is shown in Fig. 2. The data shown in Fig. 2 leads to several important observations concerning the start of self-ignition (α_{psp}) of the fuels that were tested:

- at low engine loads, only the TM/135/18 fuel has a self-ignition start value different than the other fuels. The earlier self-ignition of that fuel is correlated with a lower concentration of NO_x in the exhaust gas compared to the TM/134/18 fuel. At this engine load, the TM/135/18, as the only one of the fuels with special additives that were tested, makes it possible to reduce the NO_x concentration compared to the reference fuel; at the load of 100 Nm, the difference is as large as 5%;
- at medium engine loads the differences in the values of α_{psp} are not very large, but one can notice that, similar to the concentration of nitrogen oxides in the exhaust gas, the values of α_{psp} are monotonously decreasing for the successive fuels (TM/134/18, TM/135/18, TM/136/18, TM/154/18), which indicates an earlier start of self-ignition;
- at high engine loads, all fuels with a special additive package (at different concentrations) are characterized, also monotonously, by an ever earlier start of self-ignition; however, the changes of α_{psp} are much larger than at medium engine loads. This is probably due to the aforementioned stronger action of one of the additives at the higher temperatures in the cylinder of an engine working under a higher load. This results in a reduction of the NO_x concentration in the exhaust gas of the engine supplied with the TM/154/18 fuel and working in a higher load range, which is much greater than at medium engine loads, compared to the TM/134/18 fuel.

The impact of the analyzed fuels on the start of fuel self-ignition (in relation to the concentration of nitrogen oxides in the exhaust gas) is important because it has a direct impact on self-ignition delay (t_{sp}) – Fig. 3.

A comparison of the values of self-ignition delay, t_{sp} , for the three fuels with the content of 10% of RME with the special additive package (TM/135/18, TM/136/18, TM/154/18) with the values of t_{sp} for the reference fuel (TM/134/18) with the value of 7% of RME, leads to the following conclusion:

- The TM/135/18 fuel is characterized by a much shorter self-ignition delay compared to the TM/134/18 fuel (as much as 23%) for a low engine load. Also, when compared to the TM/136/18 and TM/154/18 fuels, at low engine loads, the self-ignition delay is the shortest in the case of the TM/135/18 fuel. This shows an excellent correspondence to the smallest NO_x concentration in the exhaust gas of the engine working under a low load and supplied with the TM/135/18 fuel, compared to the remaining fuels (Fig. 3). For example, at the engine torque equal to $Mo = 100$ Nm, the NO_x concentration in the exhaust gas of an engine burning the TM/135/18 fuel is as much as 5% smaller than the concentration for the reference fuel (TM/134/18). Unfortunately, at medium and high engine loads, there was no such beneficial impact of the fuel additives in the TM/135/18 fuel on a reduction of the self-ignition delay and the concentration of NO_x in the exhaust gas, compared to the TM/134/18 fuel. This is important because at high engine loads (when the beneficial impact of the TM/135/18 fuel is negligible), the engine emits the largest quantity of NO_x and, unfortunately, at this operating range of the engine,

the EGR exhaust gas recirculation system (which is intended to reduce the concentration of nitrogen oxides in the exhaust gas) is no longer working. Consequently, use of this type of fuel in a diesel engine is not significantly justified.

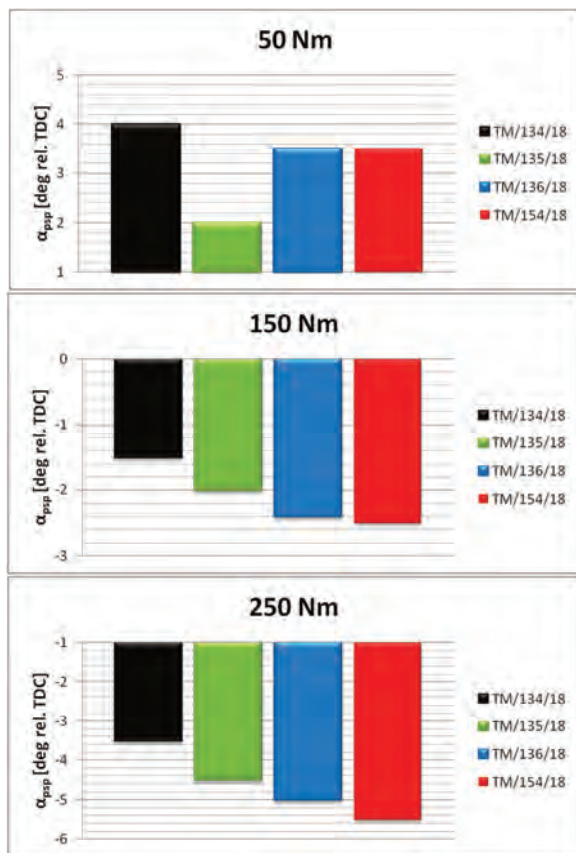


Fig. 2. Start of fuel self-ignition (α_{sp}) and NO_x in the exhaust gas at selected duty points of the engine for the four tested fuels

- The TM/136/18 fuel indicates a completely different nature of changes in the self-ignition delay (τ_{sp}), as a function of engine load, than the TM/135/18 fuel. The data shown in Fig. 3 clearly indicates that, in the entire engine load range, the self-ignition delay is shorter for this fuel than for the reference fuel TM/134/18. What is important is that the beneficial differences between the τ_{sp} values for the TM/134/18 fuel and the TM/136/18 fuel increase as the engine loads increases. For the latter fuel, the larger the engine load (higher temperature in the engine cylinder), the greater the reduction of the self-ignition delay compared to the reference fuel (TM/134/18), which corresponds to a greater reduction of NO_x content in the exhaust gas compared to the TM/134/18 fuel. Consequently, the nature of the changes (reduction of the self-ignition delay) as a function of engine load, between the TM/134/18 fuel and the TM/136/18 fuel, is very correct. However, the quantitative differences in the self-ignition delay values for the TM/134/18 fuel and the TM/136/18 fuel are small (0.5° crankshaft rotation angle) for a low engine load and 0.9° crankshaft rotation angle for a medium and high engine load). This translates into minimum differences in NO_x

concentration in the exhaust gas of an engine supplied with the TM/134/18 fuel and the TM/136/18 fuel (1 ppm for a low engine load, 2 ppm for a medium engine load, and 15 ppm for a high engine load). Those differences, especially for the low and medium engine load, are within the measurement error. For this reason, the tests were performed on the TM/154/18 fuel, which contains a two times higher concentration of an additive that reduces the self-ignition delay compared to the TM/136/18 fuel.

- The TM/154/18 fuel, as a function of engine load, leads to the same changes in self-ignition delay compared to the reference fuel (TM/134/18) as the TM/136/18 fuel, which means that at a small engine torque, the reduction of the τ_{sp} for the TM/154/18 is smaller (compared to the TM/134/18 fuel) than at a high engine load. Nevertheless, quantitatively, the reduction of the self-ignition delay for the TM/154/18 fuel is almost twice larger than for the TM/136/18 fuel. Consequently, an increase of the concentration of the additive that reduces the self-ignition delay in the TM/154/18 fuel has given the expected effect. Consequently, the concentration of NO_x determined for the fuel in question also decreased, both compared to the TM/136/18 fuel (for a high engine load by approx. 3%) and, especially, compared to the TM/134/18 fuel (for $M_o = 280$ Nm, the difference of nitrogen oxides concentration was equal to approx. 7%). This is important because such beneficial differences (of τ_{sp} and NO_x) occur at high engine loads, where the EGR system cannot be used.

The action of the fuel additives that shorten the self-ignition delay is very important because in diesel fuels the emission of nitrogen oxides is related most of all to the maximum combustion temperature (and, to a smaller degree, with the availability of oxygen and the duration of the nitrogen oxidation reaction). Consequently, the shorter the self-ignition delay (less time for gathering of the fuel dose in the combustion chamber until self-ignition), the smaller the fuel dose that self-ignites. This results in a less dynamic combustion process (smaller maximum kinematic combustion speed) and, consequently, a lower maximum combustion temperature, which leads to a lower concentration and emission of NO_x in the exhaust gas. This phenomenon is presented in Fig. 4.

On the other hand, a reduction of the self-ignition delay, τ_{sp} , of the fuel leads to a reduction of the time of mixing of fuel with air before the fuel self-ignition, which results in incomplete combustion, which in turn increases the emission of particulate matter and smoke opacity of the exhaust gas. At the same time, a reduction of the τ_{sp} shortens the time of gathering of fuel in the combustion chamber before self-ignition, which, as has been mentioned, reduces the maximum and average combustion temperature. A reduction of the combustion temperature leads to combustion of a smaller quantity of soot and PM that have been produced. Consequently, all in all, a reduction of the self-ignition delay of the fuel involves an increased quantity of smoke opacity of the exhaust gas and an increased emission of particulate matter.

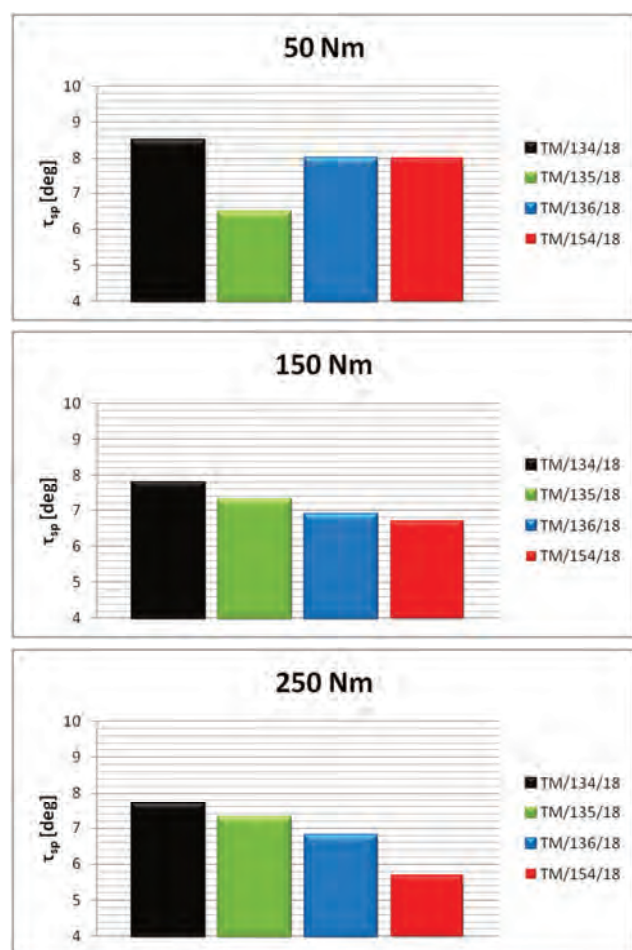


Fig. 3. Fuel self-ignition delay (τ_{sp}) and NO_x in the exhaust gas at selected duty points of the engine for the four tested fuels

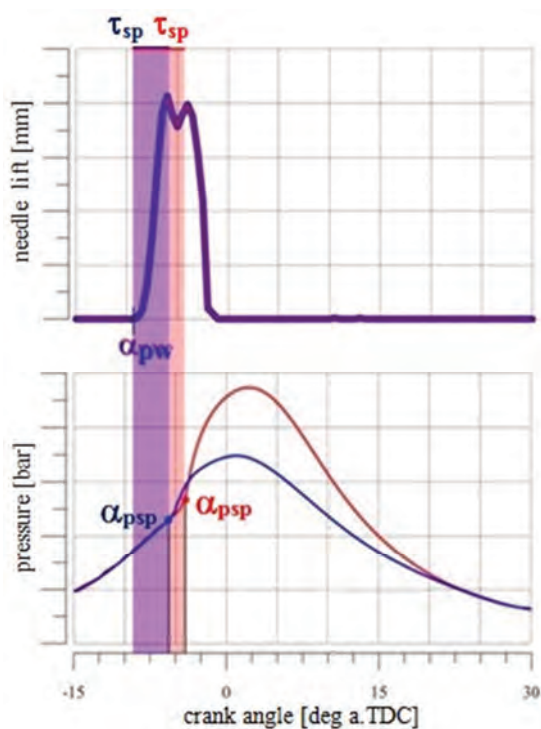


Fig. 4. Changes in the pressure in the engine cylinder as a function of the crank angle for 2 fuels: low LC value (red line), high LC value (blue line)

Usually, a reduced combustion dynamic, which is associated with a shorter self-ignition delay (τ_{sp}) results in a smaller maximum combustion pressure. The maximum combustion pressure ($p_{s,max}$), shown in Fig. 5, is related, physically, with the maximum combustion temperature, which in turn is the main cause of formation of nitrogen oxides in the engine cylinder. The lowest value of maximum combustion pressure was obtained when the engine was supplied with the TM/154/18 fuel (for each of the engine loads applied), which is greatly correlated with the measured values of concentration of nitrogen oxides in the exhaust gas of the engine.

Of note is the fact that different parameters associated with the engine being supplied with the tested fuels, determined using different measurement methods, lead to the same consistent conclusions. The TM/154/18 fuel, for which the lowest NO_x concentration was measured in the exhaust gas (especially at a high torque), is also characterized by the earliest start of self-ignition, the shortest fuel self-ignition delay, the smallest rate of pressure increase, and the maximum combustion pressure. A similar conformity was observed when comparing the TM/135/18 fuel to the TM/134/18 fuel and the TM/154/18 fuel to the TM/136/18 fuel. This indicates that there is no randomness in the tests and in their analysis, both with regard to the engine load and, especially, between the tested fuels.

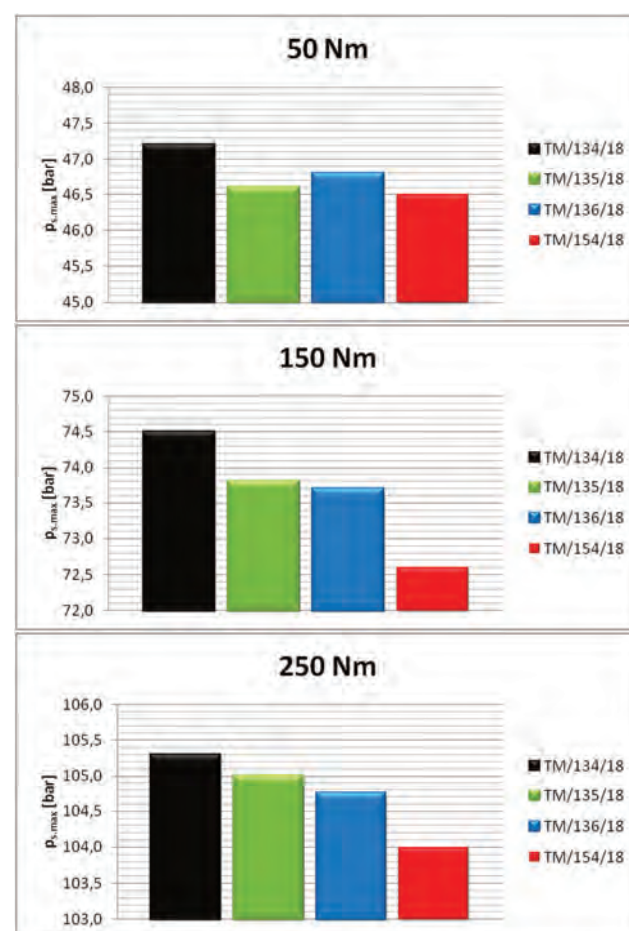


Fig. 5. Maximum combustion pressure ($p_{s,max}$) at selected duty points of the engine for the four tested fuels

4.2. Parameters of heat release

In the past, the heat release speed was calculated and analyzed much later than indicator diagrams (when the most important criterion for optimization of combustion engines was no longer general efficiency but rather the composition of the exhaust gas). The heat release speed is most often calculated based on an open indicator diagram, using different thermodynamic computational methods, with the model often used in Europe being integrated in the AVL company software [5].

Because the heat release speed depends, among others, on the crankshaft rotation angle, the engine load, and the type of fuel used, then, line in the case of open indicator diagrams, it is much easier to analyze the commonly defined parameters of those curves than the curve of a function of multiple variables. For the purpose of this paper, the following parameters were used, which were determined based on the $dQ/d\alpha$ curve:

- maximum kinetic rate of heat release $(dQ_k/d\alpha)_{\max}$,
- maximum diffusive rate of heat release $(dQ_d/d\alpha)_{\max}$,
- kinematic combustion duration $(\alpha_{sp,k})$,
- diffusive combustion duration $(\alpha_{sp,d})$.

In diesel engines of older generations (with early start of injection and without division of the fuel dose into parts), the maximum kinematic combustion speed $(dQ_k/d\alpha)_{\max}$ was significantly higher than the maximum diffusive combustion speed $(dQ_d/d\alpha)_{\max}$. In such a situation, engines emitted large amounts of both nitrogen oxides and particulate matter. A reduction of the type test limits of NO_x and PM emission resulted in design, regulation, and fuel changes (including post-process ones) in modern diesel combustion engines. Those changes involve a reduction of the maximum kinematic speed value (in order to reduce NO_x emissions) and an increase in the maximum diffusive combustion speed (in order to reduce PM emissions and smoke opacity of exhaust gas).

The brief information presented above and the title of the paper indicate that the impact of the tested fuels on the maximum value of the heat release speed in the kinetic combustion phase $(dQ_k/d\alpha)_{\max}$ is the essence of the tests presented herein. The impact of the three fuels containing a special packet of additives (TM/135/18, TM/136/18, and TM/154/18), compared to the reference fuel (TM/134/18) on the maximum kinetic combustion speed is presented in Fig. 6. The data indicates that the quantitative changes $(dQ_k/d\alpha)_{\max}$ for the analyzed fuel (for all engine loads) correspond to both the changes in the self-ignition delay (τ_{sp}) , the maximum speed of increase of the pressure in the engine cylinder $(dp/d\alpha)_{\max}$, the maximum combustion pressure $(p_{s,max})$, and of course the concentration of nitrogen oxides (NO_x) in the exhaust gas.

One must keep in mind that such parameters as e.g. hourly fuel consumption (G_{pal}), the exhaust gas temperature (t_{sp}), and the charging pressure (p_d) are the results of measurements associated with completion of a large number of engine operating cycles (and are averaged for all cylinders of the engine). On the other hand, indicator diagrams and heat release speed diagrams apply to a specific individual operating cycle of the engine (and one cylinder of the engine). Because, as has been found, the maximum kinetic

combustion speed $(dQ_k/d\alpha)_{\max}$ is the lowest among all the tested fuels, for medium and high engine loads (which are the most important from the operational point of view) for the TM/154.18 fuel, then in combination with the smallest values of t_{sp} and, most of all, the lowest value of NO_x concentration in the exhaust gas, this fuel is certainly preferable for use in diesel engines.

Because successive operating cycles of a piston combustion engine are clearly subject to natural fluctuation, then during an analysis of the fast-changing phenomena taking place in the engine cylinder, one cannot afford to analyze a randomly selected engine operating cycle. Therefore, in the tests that were performed – in the measurements and analyses of the open indicator diagrams and in the heat release speed calculated based on those diagrams for each engine duty point, a representative indicator diagram was determined which (in accordance with the research method most commonly used in the world) is the result of averaging of a hundred successive engine operating cycles. Consequently, it is possible to correlate parameters measured in many engine operating cycles (e.g. G_{pal} , t_{sp} , p_d) with the parameters of representative indicator diagrams and head release speed diagrams. This is another confirmation of the fact that the analyzed differences in the values of the parameters for the tested fuels are not random but rather result directly from the quality and quantity of the fuel additives used in the packages (TM/135/18, TM/136/18, and TM/154/18).

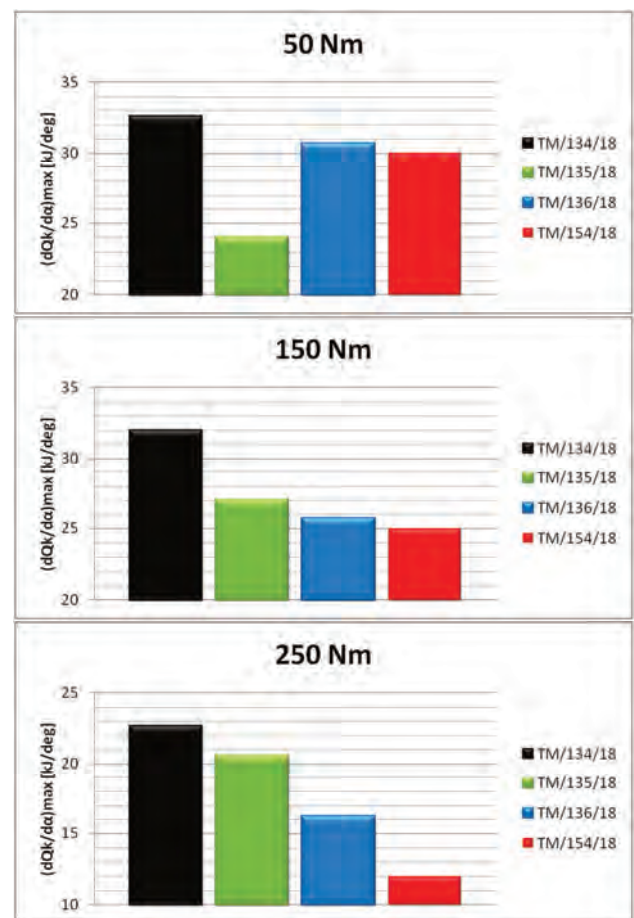


Fig. 6. Maximum heat release speed $(dQ_k/d\alpha)_{\max}$ in the engine cylinder in the kinetic combustion phase at selected engine duty point for the four tested fuels

Only the TM/135/18 fuel (compared to the TM/134/18 fuel) is characterized by a reduced maximum kinetic combustion speed (26.4%) at low engine load. This is accompanied by a significant reduction of the self-ignition delay (23.5%) and, consequently, a reduction of the concentration of nitrogen oxides in the exhaust gas.

After the kinetic combustion phase (immediately after the self-ignition of the fuel), the diffusive combustion phase takes place. The speed of diffusive combustion is associated in practice with the speed of the fuel injected into the engine cylinder (because this stage of fuel combustion occurs constantly at a high excess of air and involves a fuel-air mixture that is preliminarily mixed in the combustion chamber in the self-ignition delay period). Therefore, by controlling (increasing) the speed of fuel injection in this combustion stage, it is possible to influence the maximum diffusive combustion speed $(dQ_d/d\alpha)_{\max}$. This is important because an increase in the diffusive combustion speed usually leads to combustion of a part of soot that has been produced, which constitutes a significant part of the weight of the particulate matter (PM) and consequently reduces the overall PM emission and the smoke opacity of the exhaust gas. The maximum value of the heat release speed in the diffusive combustion phase $(dQ_d/d\alpha)_{\max}$ for 3 engine loads and the tested fuels is shown in Fig. 7.

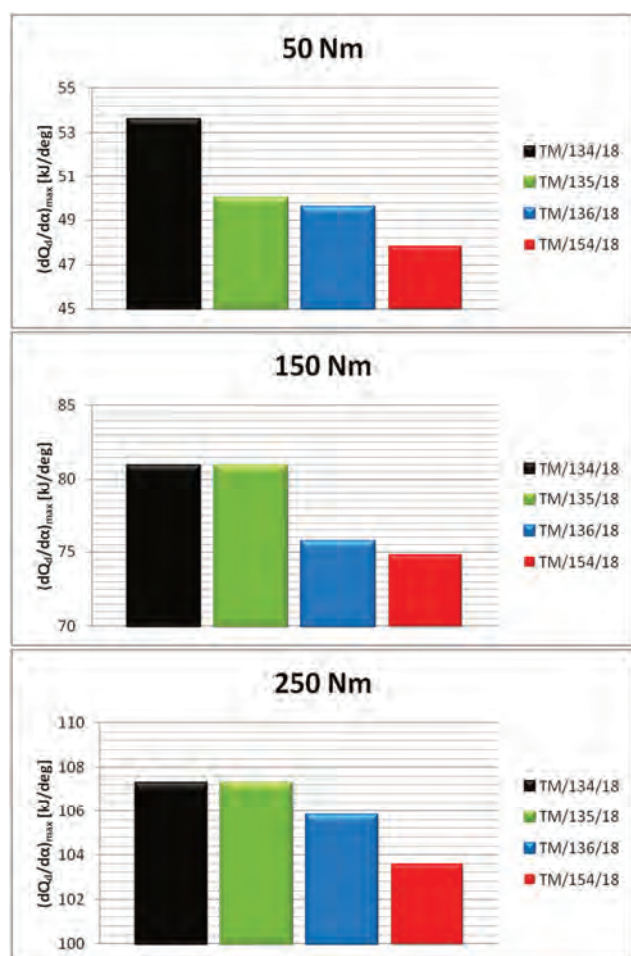


Fig. 7. Maximum heat release speed $(dQ_d/d\alpha)_{\max}$ in the engine cylinder in the diffusive combustion phase at selected engine duty point for the four tested fuels

The data indicates that in the case of the TM/135/18 fuel the maximum diffusive combustion speed $(dQ_d/d\alpha)_{\max}$ decreased only at low engine loads compared to the TM/134/18 fuel and that the former fuel had no impact (compared to the reference fuel) on the $(dQ_d/d\alpha)_{\max}$ at higher engine loads (medium and high). This involves a small (for a low engine load) reduction of the smoke opacity of the exhaust gas. This demonstrates that the additive used in this fuel (TM/135/18) has the greatest impact at this engine load and at the low temperature in the cylinder. This was not observed in the case of the TM/136/18 and TM/154/18 fuels.

Contrary to frequent preliminary analyses, a decrease of the maximum diffusive combustion speed $(dQ_d/d\alpha)_{\max}$ does not have to be accompanied by an increase in the smoke opacity of the exhaust gas. This is the case here: at a low load of the engine supplied with the TM/135/18 fuel there was a small decrease of $(dQ_d/d\alpha)_{\max}$ and, simultaneously, a reduction of smoke opacity of the exhaust gas (compared to the TM/134/18 fuel). This is possible because the reduction of the smoke opacity of the exhaust gas (D) is associated not only with the maximum value of the diffusive combustion speed but also with the duration of this combustion phase, which is often forgotten in basic analyses of the mechanism of formation of soot, which is the key cause of smoke opacity and emission of PM in exhaust gas.

However, one must keep in mind that although an increase in the value of $(dQ_d/d\alpha)_{\max}$ (e.g. by using fuel additives) is a good way to reduce PM emissions in engine exhaust gas, elongation of the combustion process in this phase, which does not bring the same effect, involves a harmful elongation of the entire combustion process and a harmful (from the standpoint of overall efficiency of the engine) shift of the end of combustion to a later moment – see the diagram in Fig. 8.

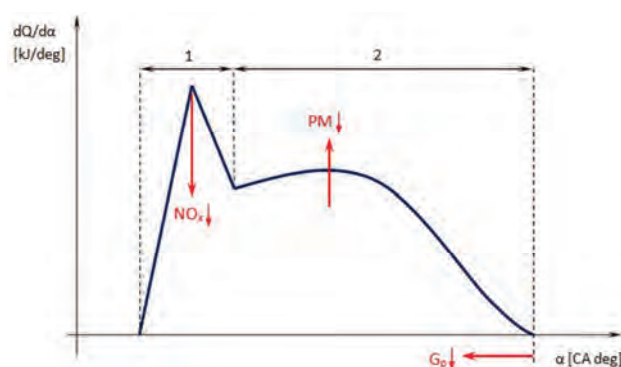


Fig. 8. Changes in the heat release speed in a diesel engine cylinder: 1 – kinetic combustion speed phase, 2 – diffusive combustion speed phase

Therefore, fuel additives should increase the combustion speed in the diffusive phase rather than extending the duration of this combustion stage. The values of the parameter defining the duration of diffusive combustion ($\alpha_{sp,d}$) are shown in Fig. 9.

The duration of diffusive combustion of the TM/135/18 fuel at low and medium engine loads is longer than the duration for the TM/134/18 fuel. Consequently, at a slightly lower maximum heat release speed in the diffusive combus-

tion phase, the much longer duration results in combustion of a larger quantity of soot that has been produced and, consequently, in a reduced smoke opacity and, most likely, a reduced emission of particulate matter in the engine exhaust gas. In the case of the TM/136/18 and TM/154/18 fuels, which have twice as high concentrations of the additive that shortens the self-ignition delay as the TM/135/18 fuel, the qualitative impact on reduction of the $(dQ_d/d\alpha)_{\max}$ (Fig. 9) is the same. However an increase of the additive that increases the value of LC in the TM/154/18 fuel (compared to the TM/136/18 fuel) resulted in both a decrease of the value of $(dQ_k/d\alpha)_{\max}$ (thanks to the shortening of the self-ignition delay t_{sp}) and, unfortunately, a decrease of the maximum diffusive combustion speed $(dQ_d/d\alpha)_{\max}$. This worsened the conditions for combustion of the soot that had been produced, which resulted in a higher smoke opacity of the exhaust gas at medium and high engine loads for the TM/136/18 and TM/154/18 fuels.

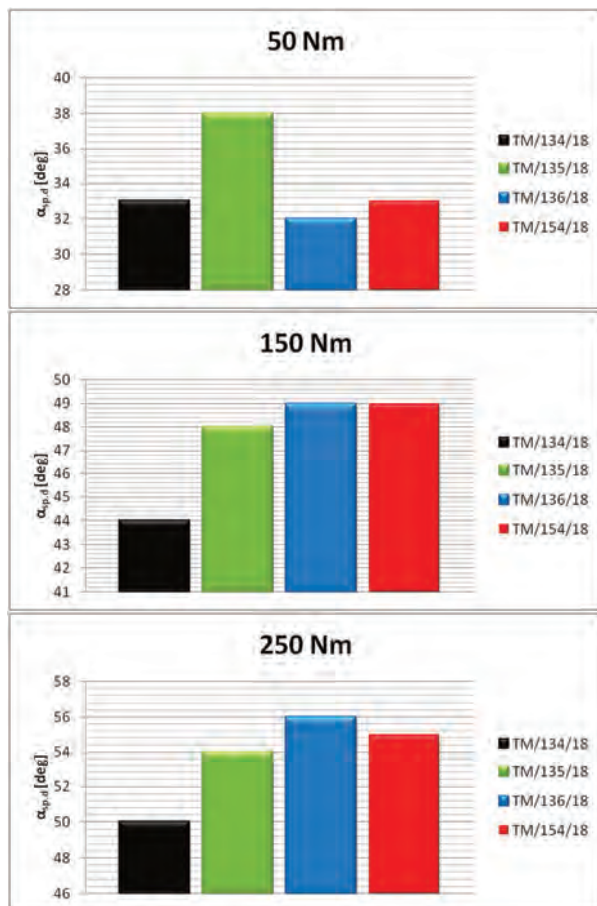


Fig. 9. Duration of diffusive combustion ($\alpha_{sp,d}$) in the cylinder at selected duty points of the engine for the four tested fuels

In the case of the TM/136/18 fuel and the TM/154/18 fuel, the process of combustion in the diffusive phase was a little longer (compared to the TM/134/18 fuel), but this did not result in a reduced smoke opacity of the exhaust gas compared to the latter fuel, because, unfortunately, of the dominant phenomenon of reduced maximum diffusive combustion speed $(dQ_d/d\alpha)_{\max}$. On the other hand, the slightly longer combustion process in the diffusive phase

for the two fuels containing a package of additives can be linked to a small increase of the hourly fuel consumption (approx. 1%) compared to the reference fuel.

The above discussion indicates that when analyzing the changes of the diffusive heat release speed in the engine cylinder one must not use only the maximum value of that parameter (e.g. when analyzing the smoke opacity of the exhaust gas and the particulate matter emission) and, instead, one should also include the duration of the process. This is important because the duration of diffusive combustion (in piston combustion engines) is many times longer than the duration of kinetic combustion ($\alpha_{sp,k}$) – Fig. 10. In the case of the TM/135/18 fuel, which is the most effective with regard to the NO_x concentration in the exhaust gas at low engine loads, thanks to the biggest reduction of the self-ignition delay and to the reduction of the maximum kinetic combustion speed, the duration of combustion in this phase does not significantly change compared to the TM/134/18 fuel. This leads to the conclusion that reduction of NO_x concentration in the exhaust gas, especially at low engine loads, is influenced only by the reduction of $(dQ_k/d\alpha)_{\max}$. The situation is different in the case of the TM/136/18 fuel and, in particular, the TM/154/18 fuel, for which a reduction of nitrogen oxides concentration in the exhaust gas is accompanied, at medium and high engine load, by a great reduction of self-ignition delay (t_{sp}), a reduction of the maximum heat release speed in the kinetic combustion phase $(dQ_k/d\alpha)_{\max}$, and, at the same time, a reduction in the duration of combustion in the kinetic phase ($\alpha_{sp,k}$).

Based on the results of the tests presented herein, one must conclude that when analyzing the heat release speed in the engine cylinder one must not focus only on the maximum value of kinetic combustion speed (compared, e.g. to the NO_x concentration in the exhaust gas) and on the maximum value of diffusive combustion speed (compared to the PM emission and the smoke opacity of the exhaust gas); instead, one should also consider the duration of combustion in the kinetic phase and the diffusive phase. Proper consideration of the value of all those parameters enables full understanding of the mechanisms of the fast-changing processes taking place in the cylinder of a piston combustion engine. One must also keep in mind that an extension of the diffusive combustion part works in two ways: on the one hand, it reduces the smoke opacity and the emission of particulate matter in the exhaust gas and, on the other hand, it increases the energy losses (which should be used to increase the internal energy of the operating medium and to perform technical work) associated with evacuation of the exhaust gas. Consequently, changes in the design, regulation, and fuel parameters in order to decrease NO_x and PM emissions should be based, most of all, on a reduction of the maximum value of the kinetic combustion speed and an increase of the maximum diffusive combustion speed, instead of on a change of the duration of those stages of the combustion process.

Conclusions

The tests that were performed made it possible to draw the following major conclusions concerning a comparison of the impact of the tested fuels (TM/135/18, TM/136/18,

and TM/154/18) with cetane-detergent additives in diesel fuel with RME content increased to 10%, compared to the reference fuel with 8% RME (TM/134/18):

1. The TM/135/18 fuel has an impact on the engine operation process basically only at low engine loads. At such loads, a shorter self-ignition delay and a lower maximum kinetic speed $(dQ_k/d\alpha)_{\max}$ were observed, associated with a small decrease of NO_x emissions in the exhaust gas. Also, at low engine loads, the maximum diffusive combustion speed $(dQ_d/d\alpha)_{\max}$ slightly decreased for this fuel. This resulted in a small (in principle within the measurement error) reduction of the smoke opacity of the exhaust gas.

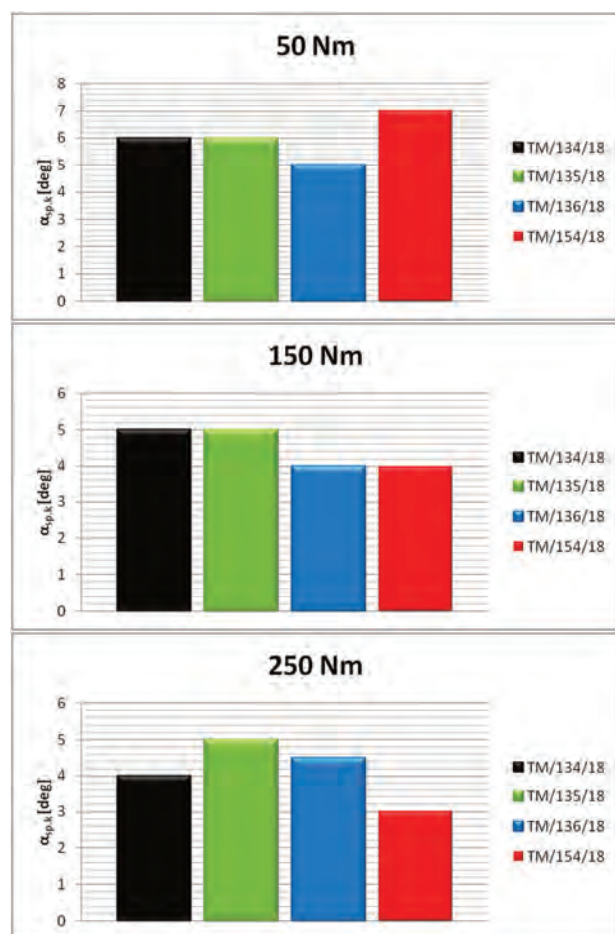


Fig. 10. Duration of kinetic combustion ($\alpha_{sp,k}$) in the cylinder at selected duty points of the engine for the four tested fuels

2. The TM/136/18 fuel has a different qualitative action compared to the TM/135/18 fuel because its impact on the operating process of the engine is noticeable only at medium and high engine loads. At such loads, in the case of the TM/136/18 fuel, the fuel self-ignition delay (t_{sp}), the maximum kinetic combustion speed $(dQ_k/d\alpha)_{\max}$, and the NO_x concentration in the exhaust gas decreased. However, because usually a decrease of the maximum kinetic combustion speed accompanies a reduction of the maximum diffusive combustion speed (a reduction of the average combustion speed), also in this case, for the TM/136/18 fuel, the maximum diffusive combustion speed $(dQ_d/d\alpha)_{\max}$ was slightly lower, which unfortunately resulted in a higher smoke opacity of the exhaust gas. As a result of the rela-

tively small change of the impact of the TM/136/18 fuel on the reduction of the self-ignition delay (t_{sp}) and on the reduction of the maximum kinetic combustion speed $(dQ_k/d\alpha)_{\max}$ and, consequently, on the small practical reduction of nitrogen oxide (NO_x) concentration in the exhaust gas, the decision was to use the TM/154/18 fuel in which the concentration of the additive that shortens the t_{sp} is twice as high as in the TM/136/18 fuel.

3. The TM/154/18 fuel is characterized by similar qualitative changes of the analyzed operating parameters of the engine as the TM/136/18 fuel, whereby the changes are much greater at medium and high engine loads. The impact of the TM/154/18 fuel, compared to the TM/134/18 fuel, for the lowest engine loads (0–80 Nm), with reference to a majority of the parameters that are important from the point of view of the energy characteristics and the composition of the engine exhaust gas, is negative. Only at higher loads (up to the maximum load), a beneficial effect was observed of the additives in the TM/154/18 fuel (whose quantity was larger than in the TM/134/18 fuel) on a significant reduction of the maximum kinetic combustion speed (by a significant reduction of the self-ignition delay) and a reduction of NO_x concentration in the engine exhaust gas (compared to the TM/134/18 fuel). Unfortunately, in such conditions, the maximum diffusive combustion speed is also reduced, which leads to an increase in smoke opacity of the engine exhaust gas at these load values (medium and high). This is a typical phenomenon associated with an attempt to control the fast-changing heat release speed in the engine cylinder: a reduction of $(dQ_k/d\alpha)_{\max}$ in order to reduce the emission of NO_x in the exhaust gas most often leads to a reduction of the average combustion speed, which also involves a reduction of the maximum diffusive combustion speed, which in turns increases the smoke opacity of the exhaust gas.

4. The use of the additive in the TM/136/18 fuel made it possible to determine the appropriate direction of quantitative changes in relation to the value of the energy characteristics and the composition of the exhaust gas (compared to the TM/134/18 fuel). However, the changes in the values of those characteristics were not satisfactory from the practical point of view because they resulted from insufficient changes in the values of the parameters that determine the physico-chemical phenomena that caused them (self-ignition delay and maximum kinetic combustion speed). This is why further tests were performed on the TM/154/18 fuel, which differed from the TM/136/18 fuel only by the greater content of the additive that reduced the self-ignition delay (t_{sp}). Compared to the TM/154/18 fuel, it must be concluded that the reduction of nitrogen oxides (NO_x) in the exhaust gas by over 3% for the load of 250 Nm and 7% for the maximum engine load, which was associated with the shorter self-ignition delay (by over 30%) and the reduction of the maximum kinetic combustion speed $(dQ_k/d\alpha)_{\max}$ compared to the reference fuel (TM/134/18) brought the expected results. Thus, it must be confirmed that only the TM/154/18 fuel has a practical beneficial effect on the significant operating parameters of the engine.

Acknowledgements

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Nomenclature

α_{pw}	start of injection	$dQ_d/d\alpha$	diffusion rate of heat release
α_{psp}	start of self-combustion	τ_s	delay of self-combustion
$\alpha_{sp,d}$	time of diffusion combustion	CI	compression ignition
$\alpha_{sp,k}$	time of kinetic combustion	$p_{s,max}$	max combustion pressure
$dQ_k/d\alpha$	kinetic rate of heat release	RME	rapeseed methyl ester

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Predesign of a cylinder head of compression ignition engine

The paper presents the initial design of the four-cylinder CI engine head and the analysis of the strength of the head with the use of FEM. The article covers: general assumptions of the designed head, analytical design calculations, three-dimensional head model and timing system components using CAD. The scope of calculations using the head calculation model includes strength calculations and determination of the amount of strain caused by the given load.

Key words: cylinder head, stress analysis, FEM

1. Preparation of 3D models of being designed cylinder head and components of timing system, elaboration of design documentation and documentation for further development of the cylinder head model

The 3D model of being designed cylinder head together with its cover (called as cam carrier) was made with use of the Autodesk Inventor Professional program, according to adopted earlier assumptions and performed analytical calculations, having simultaneously in mind effect of ordering of individual components of timing system on geometry of the structure. 3D models of timing system components have been also generated. Possibility of generation of accurate model of being designed structure has been also taken into account during creation of the cylinder head model. Effective development of the cylinder head model was possible owing to the following activities [2–7]:

- adoption of type of the timing system;
- making of assumptions about method of mutual cooperation between components of timing system;
- making selection of material for the cylinder head and components for the timing system;
- making of initial analytical calculations;
- based on performed calculations, modelling of components of the timing system;
- modelling of lower plate of the cylinder head, together with positioning and modelling of holes for valve seats in area of contour of the cylinder and seat of the injector;
- modelling of inlet and outlet channels with geometry assuring correct flow of gases;
- modelling of side walls of the cylinder head and columns, which assure positioning of the screws to connect the cylinder head with the cylinder block;
- modelling of upper plate of the cylinder head (cam carrier), together with valve seats and valve spring;
- modelling of bearing system of camshafts;
- modelling of front and rear walls together with inlet and outlet oil ducts to components operated with friction;
- arrangement and modelling of holes in the cylinder head, enabling attachment with the cam carrier;
- modelling of the cam carrier.

2. Stress analysis of the cylinder head model with use of the FEM program

Stress analysis of the cylinder head was performed with use of the ANSYS R17.2 Academic Research program [1, 5].

To perform the analysis, cylinder head model was simplified (Figs 1–4) by plugging valve seats of inlet and outlet channels, and seat of injector. Moreover, components of the timing system were removed from the geometrical model.

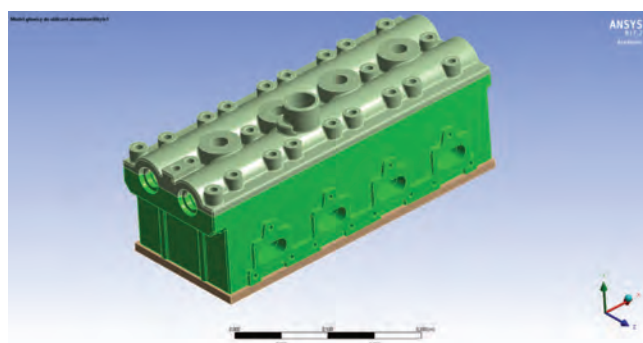


Fig. 1. Geometrical model of the cylinder head – general view

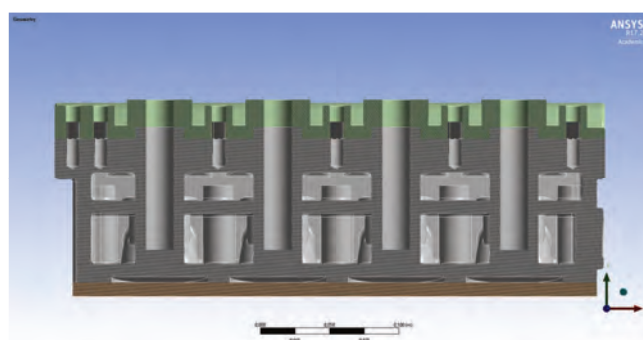


Fig. 2. Geometrical model of the cylinder head – longitudinal section

To approximate simulation to the real conditions, the model also comprises, except the cylinder head, the cam carrier (Fig. 5), which is connected (Fig. 7 and 8) to the cylinder head in locations anticipated in real structure for screw type connections. As the screw connections were not analysed in this study, they have been taken as a stiff ones, thus indicating on fact that they were suitably selected, joining the cylinder head with the cam carrier in a proper way.

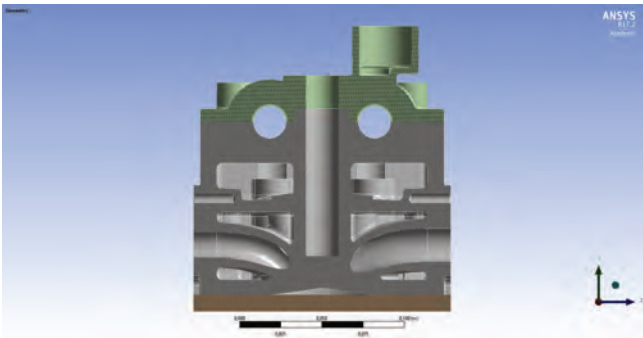


Fig. 3. Geometrical model of the cylinder head – cross section

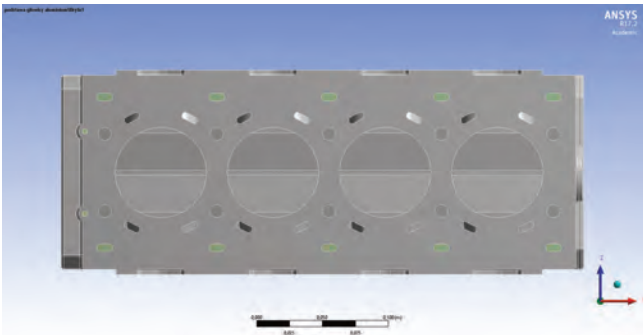


Fig. 4. Geometrical model of the cylinder head – bottom view

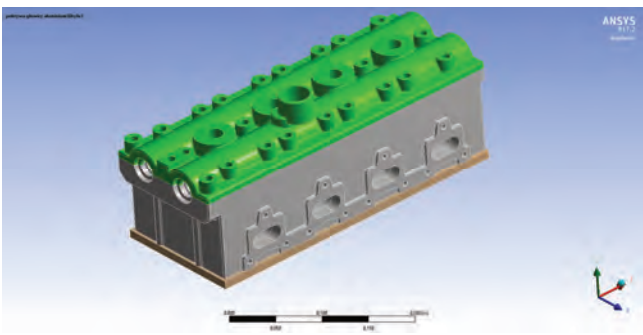


Fig. 5. Geometrical model of the cam carrier – general view

Geometrical model of the cylinder head with the cam carrier was suppressed on the plate (Fig. 6) simulating the cylinder block. Face of the plate being in contact with bottom face of bottom plate of the cylinder head was used as a support of the model. Connection of the cylinder head with the plate (Figs 7 and 8) was implemented in the same way as connection of the cylinder head with the cam carrier.

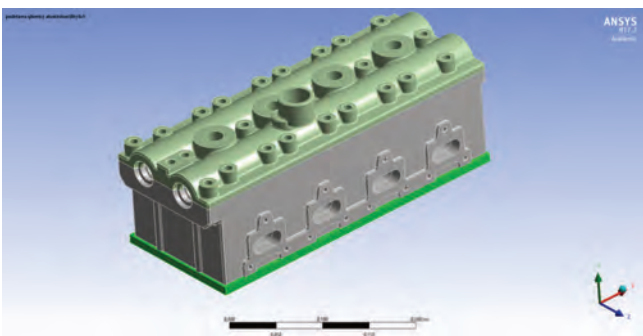


Fig. 6. Geometrical model of the bottom plate of the cylinder head – general view

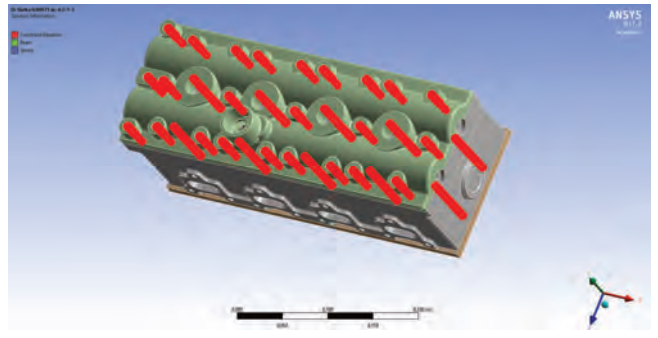


Fig. 7. Connections of the model of cylinder head with the model of cam carrier and connections of the model of cylinder head and the model of bottom plate of the cylinder head – general view

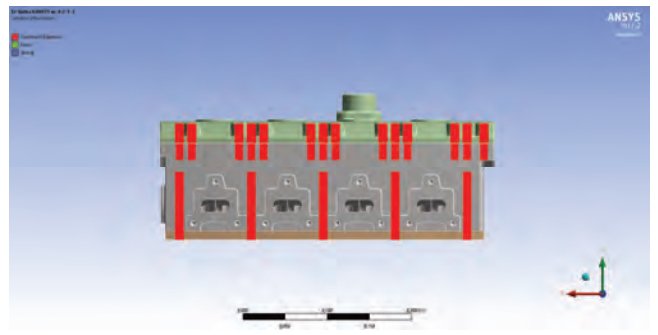


Fig. 8. Connections of the cylinder head model with the model of cam carrier, and connections of the cylinder head model with model of bottom plate of the cylinder head – view from exhaust channels side

Load of the cylinder head was applied through loading of closing surface of working space over the cylinder block of each cylinder bore separately (Fig. 9).

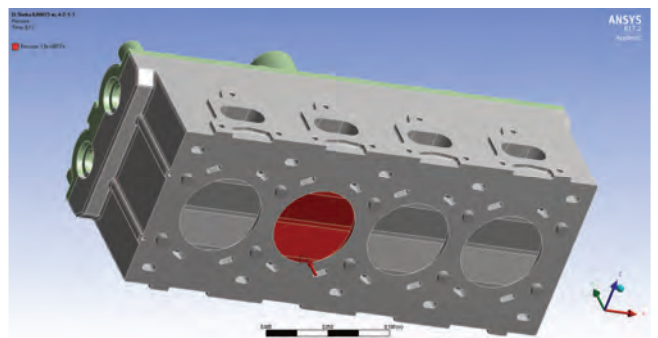


Fig. 9. Method of application of load on example of the 2nd cylinder

Imposed load was equal to maximal pressure in combustion chamber of real compression ignition engine. Read-out of its value was made on the base of indicator diagram of this engine, representing pressure course in function of crank angle (Fig. 10), in full load conditions at crankshaft speed of 4000 rpm.

Geometrical model of the cylinder head has been discretized. Equivalent to continuous system, discrete system was built from finite elements with four nodes and four walls, having edge size of 0.75 mm (Fig. 11). Such values result from limited computation power of computer system used to simulation of the model.

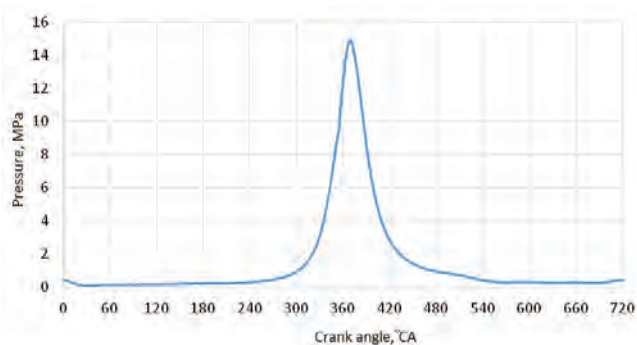


Fig. 10. Cylinder pressure run in function of crank angle of compression ignition engine for full load at 4000 rpm

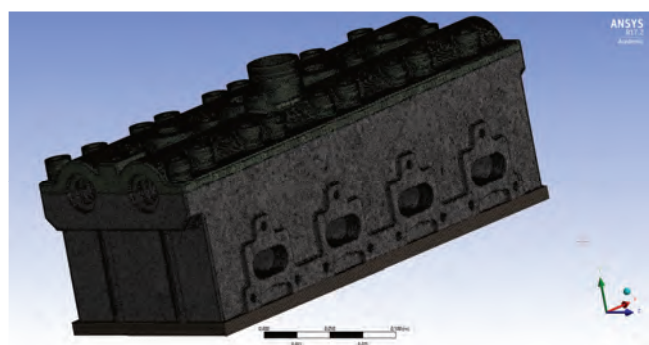


Fig. 11. Discrete geometrical model, size of finite elements equal to 0.75 mm

Calculations time amounted approximately to 30 hours for each, from four analyzed load cases.

3. Results of the investigations

In the text below are summarized obtained results of the simulations, performed with of the FEM method, which were accomplished according to the criteria adopted in the present study. Performed simulation has allowed for approximate evaluation of:

- Total deformation;
- Directional deformations in directions concurrent with „x”, „y” and „z” axes.

3.1. Simulation results of the first cylinder loading

a) Total deformations (Figs 12–15):

The highest deformations, included within limits of 0.080–0.086 mm are seen in area of valve seats, and injector seat, under load of pressure in the cylinder. Area of the

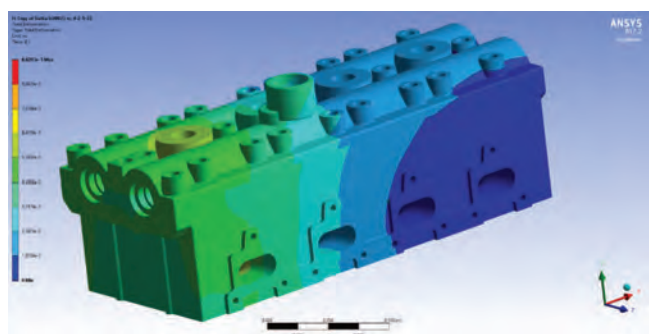


Fig. 12. Distribution of total deformations as result of simulation of the first cylinder loading– general view

deformations close to valve seats, from inlet manifold side, is larger than in area close to valve seats from exhaust manifold side.

The smallest deformations are seen in area behind the second cylinder, in direction of the third cylinder, and fall within limits of 0.000–0.010 mm.

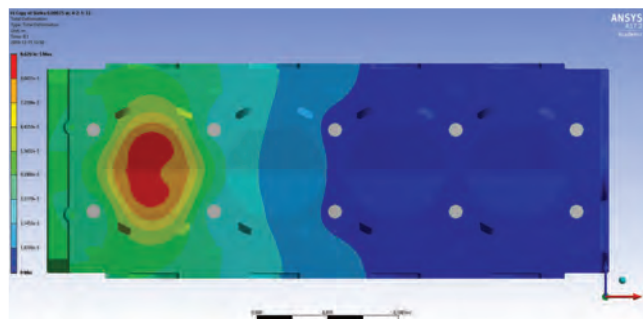


Fig. 13. Distribution of total deformations as result of simulation of the first cylinder loading – bottom view

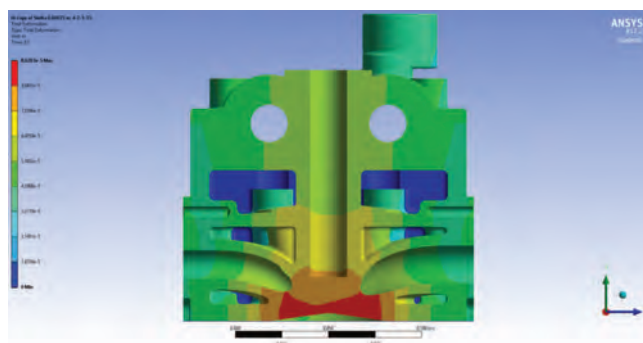


Fig. 14. Distribution of total deformations as result of simulation of the first cylinder loading – cross-section of the first cylinder



Fig. 15. Distribution of total deformations as result of simulation of the first cylinder loading – longitudinal section

b) Directional deformations concurrent with „x” axis (Figs 16 and 17):

The highest deformations in direction of „x” axis, consistent with its sense, are included within limits of 0.019 – 0.026 mm and are seen in area of oil filler in the cam carrier.

However, the highest deformations in direction of „x” axis, opposite to its sense, are included within limits of 0.003–0.013 mm, and are seen in bottom plate of the cylinder head, partially in area of loaded cylinder, and in area of the second cylinder.

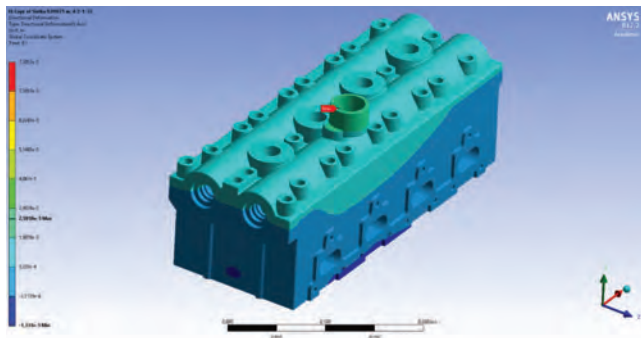


Fig. 16. Distribution of the deformations in direction of „x” axis as results of simulation of the first cylinder loading – general view

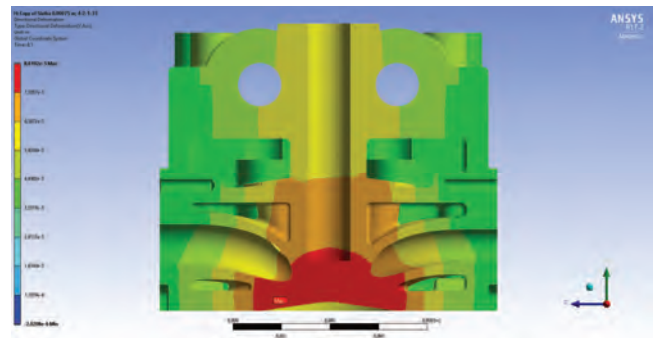


Fig. 19. Distribution of the deformations in direction of „y” axis as results of simulation of the first cylinder loading – cross-section of the first cylinder

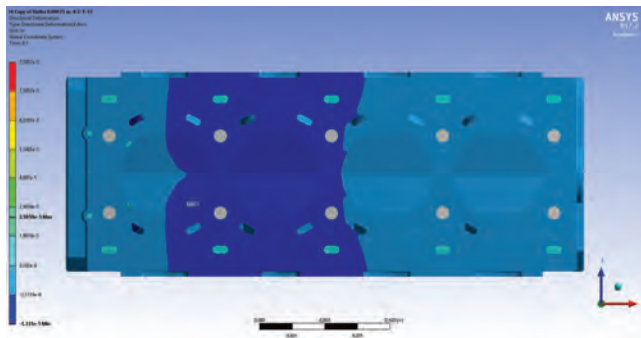


Fig. 17. Distribution of the deformations in direction of „x” axis as results of simulation of the first cylinder loading – bottom view

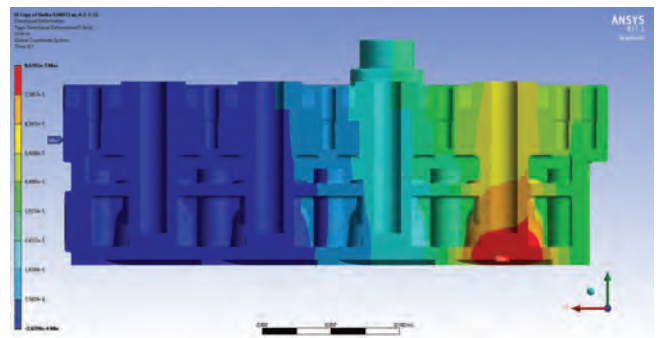


Fig. 20. Distribution of the deformations in direction of „y” axis as results of simulation of the first cylinder loading – longitudinal view

c) Directional deformations concurrent with „y” axis (Figs 18–20):

The highest deformations in direction of „y” axis, consistent with its sense, are contained within limits of 0.073–0.086 mm, and are present in area of valve seats, and in area of injector seat under load of pressure in the cylinder. Region of the deformations near valve seats, from inlet collector side, is bigger than the one near valve seats from side of exhaust channels. Moreover, deformations area is bigger for inlet valve seat, located closer to frontal wall of the cylinder head model, and the inlet channel alone from the same side.

The highest deformations in direction of „y” axis, in opposite direction to its sense, are seen in area behind the second cylinder, towards the third cylinder, and they do not exceed value of 0.002 mm.

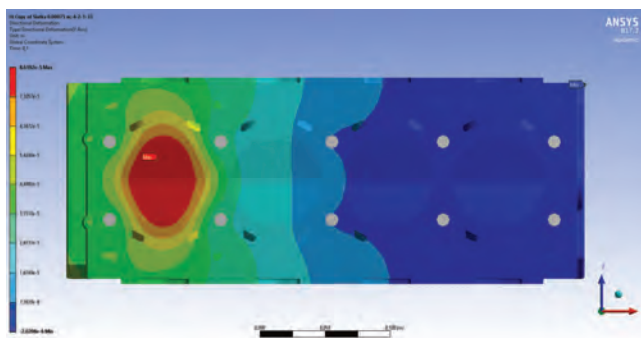


Fig. 18. Distribution of the deformations in direction of „y” axis as results of simulation of the first cylinder loading – bottom view

d) Directional deformations concurrent with „z” axis (Fig. 21):

The highest deformations in direction of „z” axis, in direction of its sense, do not exceed value of 0.009 mm and occur at external part of inlet channel wall of loaded cylinder.

In turn, the highest deformations in direction of „z” axis, opposite to its sense, do not exceed value of 0.009 mm and are present on external part of exhaust channel wall of loaded cylinder.

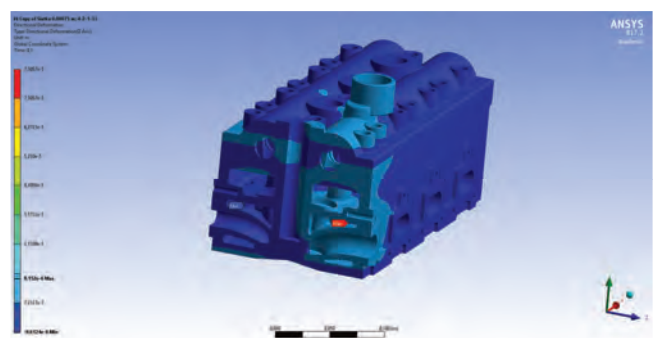


Fig. 21. Distribution of deformations in direction of „z” axis as result of simulation of the first cylinder loading – general view

3.2. Simulation results of the second cylinder loading

a) Total deformations (Figs 22–25):

The highest deformations, included within interval of 0.079–0.082 mm are present in area of inlet valve seats of loaded cylinder. Also, high deformations can be seen in area of injector seat and exhaust valve seats of the cylinder under pressure load.

The lowest deformations are present in area behind the third cylinder, in direction of the fourth cylinder, and also on frontal plate of the cylinder head, are included within range of 0.000–0.010 mm.

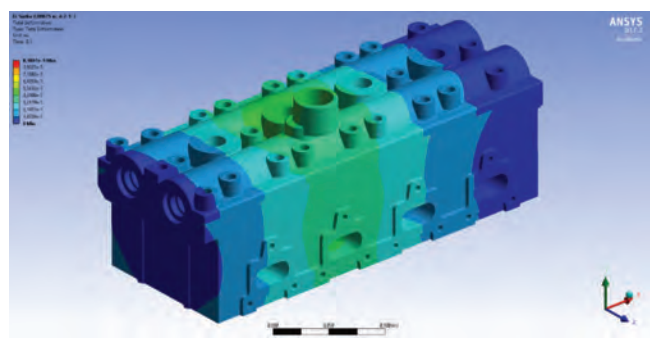


Fig. 22. Distribution of total deformations as result of simulation of second cylinder loading – general view

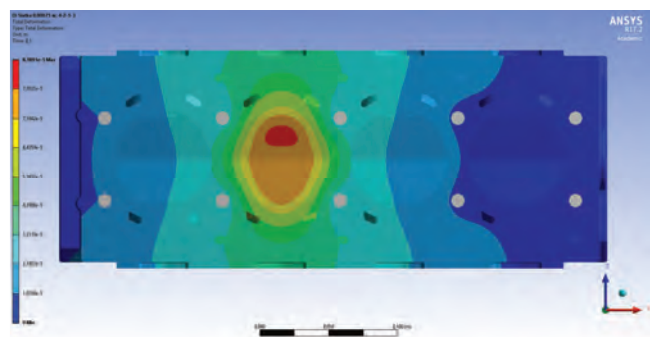


Fig. 23. Distribution of total deformations as result of simulation of second cylinder loading – bottom view

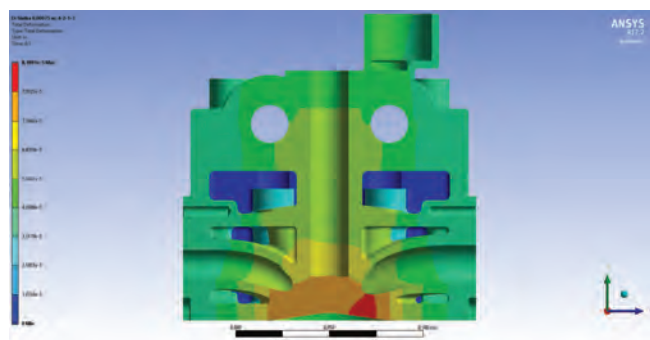


Fig. 24. Distribution of total deformations as result of simulation of second cylinder loading – cross-section of the second cylinder

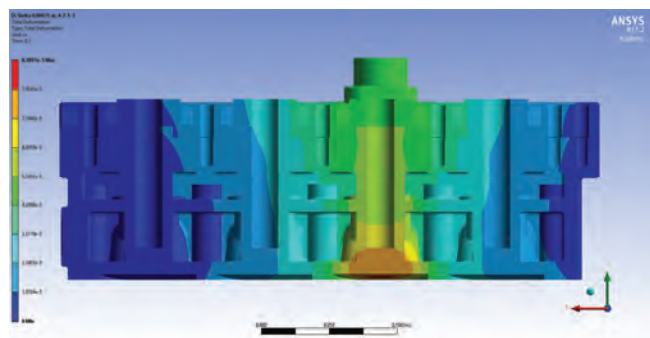


Fig. 25. Distribution of total deformations as result of simulation of the second cylinder loading – longitudinal section

3.3. Results of simulation of the third cylinder loading

a) Total deformations (Figs 26–29):

The highest deformations, contained within limits of 0.079–0.082 mm are present in area of inlet valve seats of loaded cylinder. Also high deformations can be observed in vicinity of injector seat and valve seats of exhaust channel of the cylinder under pressure load.

The lowest deformations appear in area behind the second cylinder, looking towards the first cylinder, and also in rear plate of the cylinder head, and are include with limits of 0.000–0.010 mm

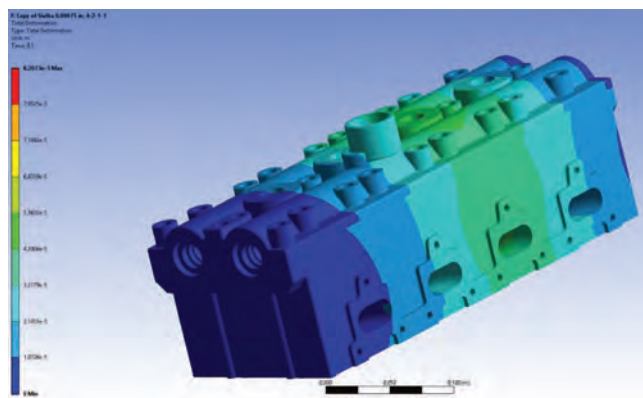


Fig. 26. Distribution of total deformations as result of simulation of the third cylinder loading – general view

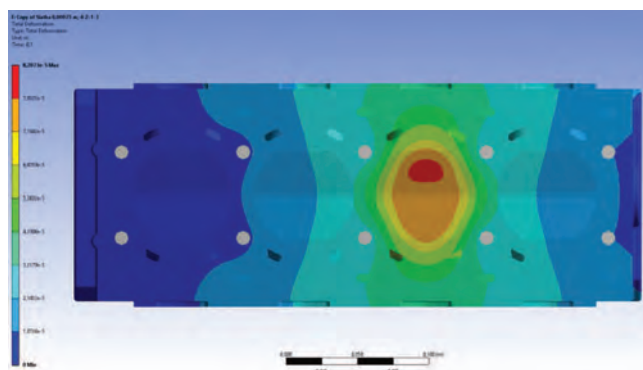


Fig. 27. Distribution of total deformations as result of simulation of third cylinder loading – bottom view

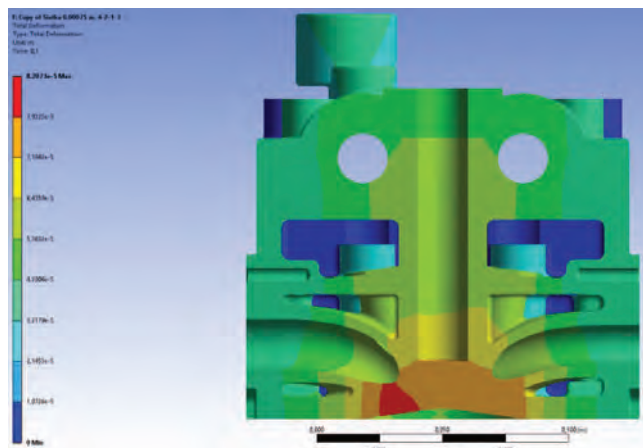


Fig. 28. Distribution of total deformations as result of simulation of the third cylinder loading – cross-section of the third cylinder

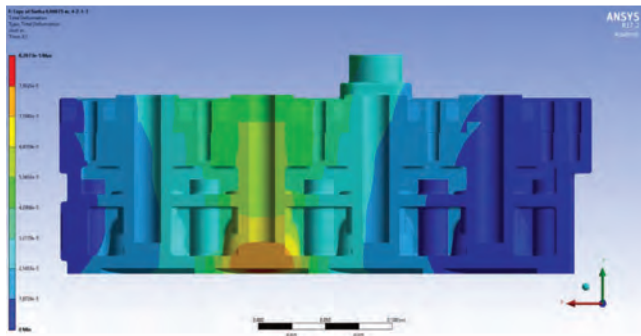


Fig. 29. Distribution of total deformations as result of simulation of the third cylinder loading – longitudinal section

3.4. Simulation results of the fourth cylinder loading

a) Total deformations(Figs 30–33):

The highest deformations, included within limits of 0.079–0.086 mm occur in area of valve seats of inlet channel of loaded cylinder, in vicinity of the rear plate. Additionally, high deformations can be seen near seat of the injector and exhaust valve seats of the cylinder under pressure load.

The lowest deformations are present in area behind the third cylinder, looking towards direction of the second cylinder, and also in rear plate of the cylinder head, and are included within range of 0.000–0.012 mm.

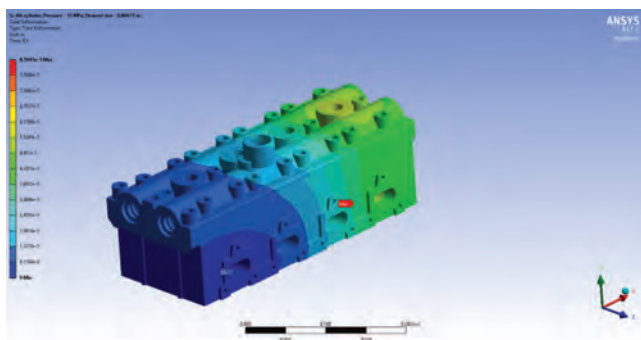


Fig. 30. Distribution of total deformations as result of simulation of the fourth cylinder loading – general view

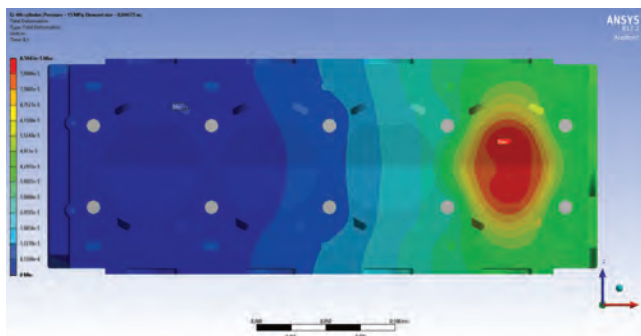


Fig. 31. Distribution of total deformations as result of simulation of the fourth cylinder loading – bottom view

4. Conclusions

- The highest total deformations are present during simulation of loading of the first and the fourth cylinder, and amount to 0.086 mm;
- The lowest total deformations occur during simulation of loading of the second and the third cylinder, and amount to 0.082 mm;

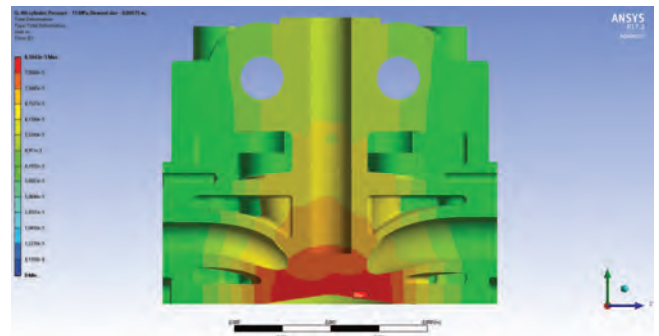


Fig. 32. Distribution of total deformations as result of simulation of the fourth cylinder loading – cross-section of the fourth cylinder

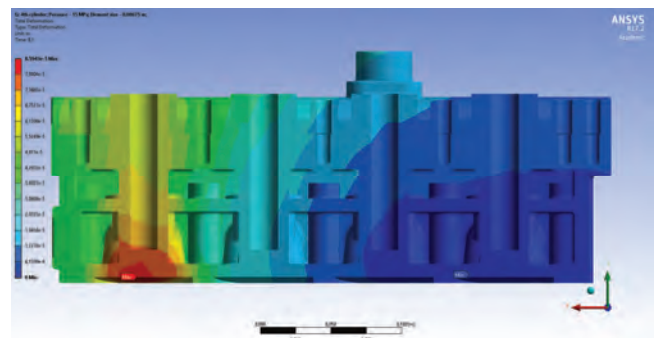


Fig. 33. Distribution of total deformations as result of simulation of the fourth cylinder loading – longitudinal section

- The highest directional deformations in direction of „x” axis occur during simulation of loading of the first cylinder, and amount to 0.026 mm;
- The lowest directional deformations in direction of „x” axis occur during simulation of loading of the second cylinder, and amount to 0.014 mm;
- The highest directional deformations in direction of „y” axis occur during simulation of loading of the first and the fourth cylinder, and amount to 0.086 mm;
- The lowest directional deformations in direction of „y” axis occur during simulation of loading of the second and the third cylinder, and amount to 0.082 mm;
- The highest directional deformations in direction of „z” axis occur during simulation of loading of the second cylinder, and amount to 0.015 mm;
- The lowest directional deformations in direction of „z” axis occur during simulation of loading of the first cylinder, and amount to 0.009 mm;
- Deformations in direction of the „y” axis have the biggest effect on the total deformations;
- The highest deformations in direction of „y” axis are seen in case of the 1st and the 4th cylinder of the cylinder head;
- In course of design work of a cylinder heads it is necessary to perform careful analysis of already existing design solutions, to make layout of individual components for which the cylinder head serves as a kind of “jointer”, and to have in mind feasibility of manufacturing of real component;
- Size of finite elements constituting mesh of discrete model is of very high importance for approximation of simulation results to result from real tests of loaded

- components. Smaller finite elements of the mesh enable obtainment of more accurate and burdened with lower error results;
- Processor capacity of a computer system performing calculations based on the Finite Element Methods is very important, because it allows for calculation of more complex cases;
 - Lack of sufficient calculation capacity of computer system accomplishing numerical simulations of the cylinder head model has resulted in lacking possibility of generation of the mesh with size of finite element smaller than 0.75 mm, and lacking possibility of simultaneous performing of the simulations according to adopted assumptions;
 - Authors of this study has in mind, that values of the deformations obtained in result of performed simulation of the cylinder head loading may differ from the real ones, and can exceed acceptable maximal deformations;
 - Both the CAD and FEM programs are very useful tools in design engineer work, facilitating and accelerating its work with simultaneous reduction of costs connected with production of many constructions, among others.
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The EMITRANSYS model and the possibilities of its application for the analysis of the development of sustainable transport systems

The transport system has a direct impact on the economic and socio-economic situation of each country. Planning its development is an important element of policy. Bearing in mind the importance and complexity of this issue, it should be considered using both quantitative methods and mathematical models, as well as advanced simulation tools for modelling transport. The article presents a proposal for an approach to simulation tests of the different transport system in the aspect of the emission of harmful exhaust gas compounds from road transport. The tool for analysing and assessing the functioning of existing or designed systems is a model in which the properties of the real system, which are important from the point of view of the research objective, should be mapped. The Model of Proecological Transport System (MEST) was presented formally as an ordered four. Also elements of optimization tasks shaping of proecological transport system in form of datas, decision variables, conditions and goal function were presented. An example of tests on simulation model was shown.

Key words: development of transport system, simulation tools, EMITRANSYS, mathematical model

1. Introduction

The transport system has a direct impact on the economic and socio-economic situation of each country [1]. Planning its development is an important element of policy [5]. Bearing in mind the importance and complexity of this issue, it should be considered using both quantitative methods and mathematical models, as well as advanced simulation tools for modelling transport. In these approaches, parameterization of models is an important element, which allows to achieve research goals with enough accuracy [6].

Typically, when analysing the operation of the different transport system, the transport subsystems of particular modes of transport are distinguished. Then, the research of these transport subsystems, the relationships between them, the impact of the environment on them and their surroundings are conducted. Interaction study of transport subsystems is an important element of modelling the transport system as a whole. These dependencies directly shape the modal split of transport tasks directed to the transport system from the environment [4]. At the same time, changes in the transport system lead to changes in the communication behaviour of residents, including changes in the demand for transport.

Summing up, modelling the development of the different transport systems, e.g. national, urban, regional, passengers and others, requires taking into account, among others:

- the needs of buyers of transport services along with their sensitivity to changes in the transport system,
- the existing and projected technical potential of transport service providers, including knowledge of restrictions on changes,
- the existing and forecasted condition of transport infrastructure, as well as investment opportunities in this respect,
- current and future solutions in the field of traffic organization in specific areas, regions,
- transport policy of a given country or region, including ecological conditions of the development of the transport system,

- the existing and projected negative impact of transport on the natural environment and the possibilities of its reduction.

Both socio-economic development and mutual cooperation of countries implies the development of transport needs, the implementation of which is carried out by various branches of transport. The division of transport tasks between particular branches of transport, especially in the conditions of a market economy, is primarily determined by the result of the economic account conducted by transport users. Until recently, the issues of transport ecology for many participants of the transport process were not significant, although many documents were issued in this area. Nowadays, the need to limit the negative impact of transport on the natural environment is a determinant of economic development.

The article presents a proposal for an approach to simulation tests of the national transport system in the aspect of the emission of harmful exhaust gas compounds from road transport.

2. The development of transport and the emission of harmful exhaust gases

Road transport is considered as the most harmful among transport branches. In 2016, road transport was responsible for almost 98% of greenhouse gas emissions emitted by transport [2, 24].

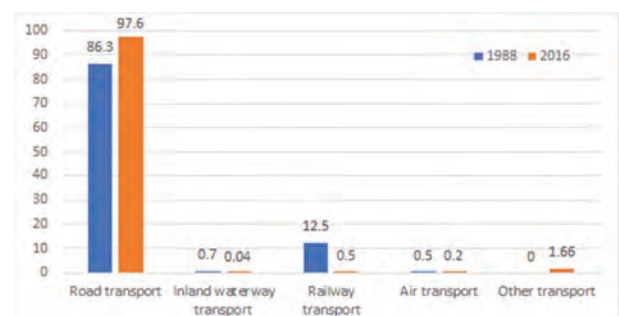


Fig. 1. CO₂ emissions from fossil fuels by sectors in Poland [18]

The negative impact of transport on the natural environment is expressed by [17]:

- greenhouse gas emissions contributing to climate change,
- local emission of air pollutants negatively affecting human health and the local natural environment,
- occupying valuable natural areas and cutting their continuity (fragmentation) with newly built sequences of technical infrastructure, contributing to the loss of biodiversity,
- noise emission threatening human health.

Air pollution emitted by vehicles depends on many factors. Among them can be distinguished: fuel composition, type and basic characteristics of vehicles, distribution of transport infrastructure, speed, places of congestion, etc. Among the air pollution measures can be distinguished: emission volume and concentration of individual primary pollutants (nitrogen oxides (NO_x), carbon monoxide (CO), sulphur dioxide (SO_2) and particulates – PM_{10} and $\text{PM}_{2.5}$, as well as dust and soot). These pollutions can have a negative impact on materials and buildings, agricultural crops and forests, and can be harmful to human health and life. Figure 2 presents changes in NO_x , CO and particulate matter emissions of $\text{PM}_{2.5}$ from road transport in Poland in recent years.

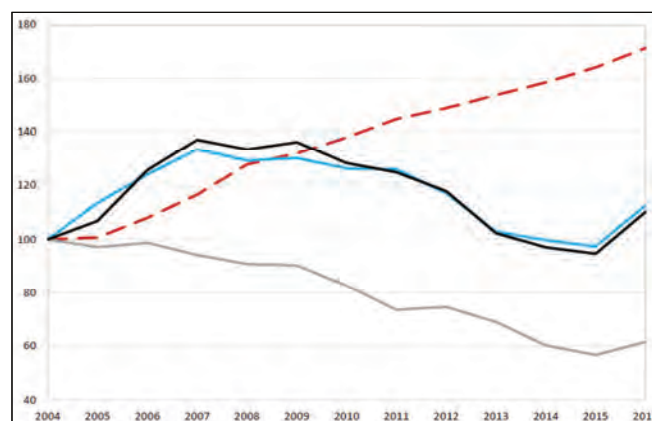


Fig. 2. Changes in emissions of NO_x , CO and particulate matter $\text{PM}_{2.5}$ from road transport in Poland in the years 2004-2016 in relation to the change in the number of cars and tractors, assuming that the volume of emissions in 2004 = 100% (Legend: red line – percentage change in the number of cars and tractors, blue line – percentage change in NO_x emissions from road transport, grey line – percentage change in CO emissions from road transport, navy blue line – percentage change in $\text{PM}_{2.5}$ emissions from road transport) [https://www.nik.gov.pl/plik/id,19075,vp,21678.pdf]

The emission of environmentally harmful substances from motor vehicles results not only from the composition and properties of fuels [21], as is the case for sulphur compounds, ring hydrocarbons and heavy metals. In addition – due to the threat of the environment from the effects of global warming – the carbon dioxide emission, which is the most important due to the scale of greenhouse gas emissions from the combustion of fuels for internal combustion engines, is considered. An additional factor related to the impact of motorization on the environment are dusts consisting of, among others, from particulates (PM_{10} , $\text{PM}_{2.5}$) (in 2011, the liner emission in Poland (connected with road transport) was responsible for 63 percent of PM_{10} concentration). At the same time, the amount of dust emissions

resulting from fuel combustion in car engines is estimated at 7%. A dozen of percent are particles from tires and brake pads. The remaining 80% results from the so-called secondary dusting, that is, from induction of dust lying on the ground in the air by passing cars. Summing up low emissions and cars mutually reinforce the phenomenon of smog.

3. Decision-making problems in shaping environmentally friendly transport systems

As mentioned before, shaping of the transport system in Poland results, on the one hand, from the forecasts of transport needs and the second from adapting the infrastructure and requirements of law to the standards and requirements of the European Union. The European Commission pays particular attention to the need for sustainable development of transport and environmentally friendly transport. Minimizing the negative impact of transport on the natural environment, and at the same time meeting expectations as to transport capacity, seem to be opposite objectives. It is therefore necessary to look for a solution that satisfies these positions.

Despite the fact that activities to reduce the negative effects of transport are constantly being taken, the continuous increase in the number of carried people and loads causes the effects of these activities are reduced to a large extent by the increased transport performance of both freight and passenger transport.

Summing up the planning of the development of the national transport system taking into account the aspects of environmental protection, it is not only related to the operational objectives of transport policy, but also to environmental issues such as air, soil and water pollution, noise or vibrations emitted by means of transport [19, 20].

The pollutant emission test is a complex process and requires the use of specialized equipment. There are various methods of conducting research. The most general classification is the division into tests in laboratory conditions and in real conditions. In the developed simulation model, the results of research on the emissivity of road transport in real conditions were taken into account as part of the EMITRANSYS project. These tests identified functional dependencies of the actual emission of particular harmful exhaust compounds from the parameters of the traffic flow, including vehicle speeds, as well as the type of transport means (capacity/load of the vehicle, Euro class, type of fuel).

The pursuit of a sustainable transport system is based on its appropriate shaping and organization in a way that allows achieving such a method of meeting the demand for transport due to the division of resources, for which it will limit the difficulties in users access to the transport system while reducing the congestion of the system and its excessive nuisance to the environment. To achieve this goal, IT tools are used that enable modelling of transport systems and then running on the obtained experimental models.

Due to the importance of transport infrastructure in meeting transport needs of passengers and transport of goods in the model, national and voivodship roads as well as railway lines and operated inland and aerial routes were taken into account. Transport networks of individual types of transport have been imposed on each other and connected in appropriate nodes, thus enabling the change in modes of transport to be included in the simulation model [12].

Due to the adopted detail of the EMITRANSYS simulation model, it was necessary to aggregate stops located in the area of one transport node. Moreover, in the simulation model, local communication lines identified as intra-district were omitted.

4. Decision model for sustainable transport systems

4.1. General assumptions

When analysing or assessing any phenomenon, process or system, the point of view from which this analysis or assessment will be carried out should always be specified. In the case of a sustainable transport system, the assessment indicator may be, for example, emission levels, external costs [9], quality of tasks, etc.

Considering the above, the following types among elements of a sustainable transport system have been distinguished:

- transport origin points,
- points of change of road characteristics and change of the type of transport means together with the accompanying point infrastructure of transport,
- transport destination points,
- transport links between the above-mentioned points appearing as existing linear transport infrastructure,
- means of transport, including economic parameters and characteristics of harmful compounds emitted,
- organization and communication networks.

The tool for analysing and assessing the functioning of existing or designed systems is a model in which the properties of the real system, which are important from the point of view of the research objective, should be mapped [8]. Formally, the Model of Proecological Transport System (MEST) was recorded as an ordered four in the following forms [10, 11]:

$$\text{MEST} = \langle \text{GE}, \text{FE}, \text{QE}, \text{OE} \rangle \quad (1)$$

where: **MEST** – model of a proecological transport system, **GE** – structure of a proecological transport system, **FE** – set of characteristics of proecological structure elements of the transport system, **QE** – volume of transport tasks, which MEST should realize, **OE** – organization.

Due to the dominating share of road and rail transport in the transport of passengers and goods in Poland, in the MEST mathematical model only these two branches of transport were taken into account (there is the possibility of extending the model with other transport modes) [23]. The transport network was identified and then analysed analytically, i.e. elements of point and linear transport and logistics infrastructure.

4.2. Parameters of the model

The MEST model also includes the characteristics of road infrastructure, means of transport and parameters of harmful exhaust gas compounds [14]. Elements of the analysed system were formally recorded. The most important of the used signs are listed below:

i, i', i''	index of the transport node, $i, i', i'' \in \text{WE}$
st	index of the type of transport mean, $st \in \text{ST}$
rsp	engine type index, $rsp \in \text{RSP}$
s	exhaust type index, $s \in \text{S}$

o	area type index, $o \in \text{O}$
neu	Euro standard index, $neu \in \text{NEU}$
v	speed index, $v \in \text{V}$
k	index of transport type (railway, road, air, ...), $k \in \text{K}$
p	road index in relation of carriage, $(a,b) \in \text{E}$, $p \in \text{P}^{ab}$
m	index of the transport kind (e.g. passenger), $m \in \text{M}$
$nr(st)$	number of euro standard for vehicle type st ,
$l(i,i')$	length of (i,i') transport connection,
$x(a,b,m)$	demand for transport in relation $(a,b) \in \text{E}$ of the m kind of transport,
$q(st,m)$	the average use of the load/capacity of the vehicle type st of m kind of transport,
$d(i,i')$	capacity of (i,i') transport connection,
$\chi(k)$	economic efficiency of transports made by k transport subsystem,
$\mu(k)$	economic efficiency of transport work performed by k transport subsystem,
$em(s,st,n$ $eu,rs,v)$	emission of harmful exhaust gas s by type of vehicle st with engine type rsp comply with neu emission standard when driving at a speed v per unit distance,
$neu(i,i')$	number of the acceptable euro standard for the section $(i,i') \in \text{LE}$,
$q(rt,k)$	level of transport carried out by k transport subsystem rt type of transport,
$ES(s,k)$	total emission of the harmful component of exhaust gas through k transport subsystem,
$L(rt,k)$	average transport distance under k transport subsystem for rt type of transport,
$KS(k)$	cost of operation k transport subsystem.

4.3. Boundary conditions and criteria for the assessment of the solution's quality

As the basic criteria for the assessment of distribution of traffic flow on transport network in the area of shaping the proecological transport system, the economic as well as ecological and social indicators were taken into account in the developed model. One of the most important criteria for assessing the distribution of a traffic flow in the model of proecological transport system are the emission levels of individual harmful exhaust gas emissions emitted by transport means performing transport tasks in a selected area of the transport network:

$$\begin{aligned} \forall s \in \text{S} \quad \forall k \in \text{K} \quad ES(s,k) = & \quad (2) \\ = 3600 \cdot 10^{-6} \sum_{(a,b) \in \text{E}} \sum_{p \in \text{P}^{ab}} \sum_{st \in \text{ST}(k)} \sum_{ne \in \text{NE}} \sum_{rs \in \text{RS}} xp(p,a,b,st,ne,rs) \times \\ & \times \sum_{(i,i') \in \text{LE: } st \in \text{ST}(i,i') \wedge p \in \text{P}_{i,i'}^{ab}} \frac{l(i,i')}{v(i,i')} \cdot em(s,st,ne,rs,i,i') \end{aligned}$$

Next developed criterion for assessing the distribution of traffic flow on the transport network, which is economically significant, is the direct transport cost. Taking into account transport performance as a result of the functioning of the transport system, the index of economic efficiency of transport work is determined, while taking into account the level of transport – the index of economic efficiency of

transport. Assuming the possibility of decomposing the operating costs of the transport system to the costs of the passenger transport subsystem and freight transport, the analytical form of these indicators was recorded in the form of:

$$\forall k \in K \quad \mu(k) = \frac{\sum_{rt \in RT} q(rt, k) \cdot L(rt, k)}{KS(k)} \quad (3)$$

$$\forall k \in K \quad \chi(k) = \frac{\sum_{rt \in RT} q(rt, k)}{KS(k)} \quad (4)$$

The model includes, among others [13]:

1. constraints on the fulfilment of transport needs,
2. constraints resulting from the specified capacity of connections (sections) of roads,
3. constraints imposed on the traffic flow,
4. constraints on enter the area due to the acceptable Euro standard,
5. constraints on type of decision variable,
6. constraints on not to exceed the available number of transport means,
7. constraints on the possibility of moving transport means of particular types on transport connections,
8. constraints on the possibility of servicing routes with vehicles meeting specific emission standards,
9. and other constraints.

5. Examples of the practical application of the EMITRANSYS model

In the last part of the article the possibility of using the EMITRANSYS model for conducting analyses will be presented. The developed EMITRANSYS simulation model of the environment-friendly transport system has been divided into two interdependent models:

1. network model that maps the point and linear infrastructure of the transport system and describes the availability of system users to specific branches and means of transport. It consists of, among others, with:
 - sections of roads, railways, inland roads, air corridors etc.,
 - nodes being their destinations, origins or connection elements,
 - communication areas, nodes and public transport stops,
 - public transport sections and lines with timetables,
2. demand model containing data on current and future transport needs, including the demand for passenger travel and cargo transportation between specific areas of the network.

In example it has been assumed that the movement of goods can be implemented from any node on the transport network, to another located along the same or different communication path. For this reason, in fact, for the same route of carriage will be chosen different routes of carriage, and hence different types of transport.

One of the tools for estimating the level of environmental pollution used in the EMITRANSYS simulation model is the HBEFA module implemented in PTV VISUM. It serves to map pollutant emissions from transport network connections based on the distribution of the traffic flow, the

assumed vehicle structure and unit emission factors (including CO₂, CO, HC, NO_x, PM_x). This module contains a database that includes historical data from 1990 to forecasts for 2035. The structure of vehicles includes participation within a given category (light delivery vans, lorries, tractors with semitrailer, refrigerated vehicles), motorcycles, passenger cars, city buses and tourist buses) for different types distinguished in terms of e.g. capacity, number of seats, engine type (gasoline, diesel). This module makes it possible to determine the emission of pollutants on the transport network for a given vehicle structure.

One of the possibility of using EMITRANSYS simulation model [15] is the estimation of the emission volumes of the previously mentioned harmful substances (CO, NO_x, HC) by particular vehicle types, with a specific emission standard and with a specific engine type. Thanks to this, after the distribution of flow of passengers and loads on the transport network of Poland, the level of emission of harmful exhaust compounds was estimated on individual sections of the Polish road network [16, 22]. In addition, the emissions from passenger transport were distinguished. Obtained results for 2015, 2020 and 2025 are shown in Figs 3–5.

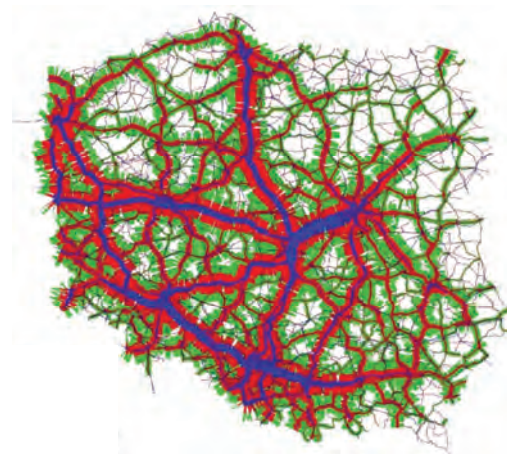


Fig. 3. The level of emissions of harmful compounds by passenger transport in 2015 in g/km (legend: blue line – HC, red line – CO, green line – NO_x) [8]

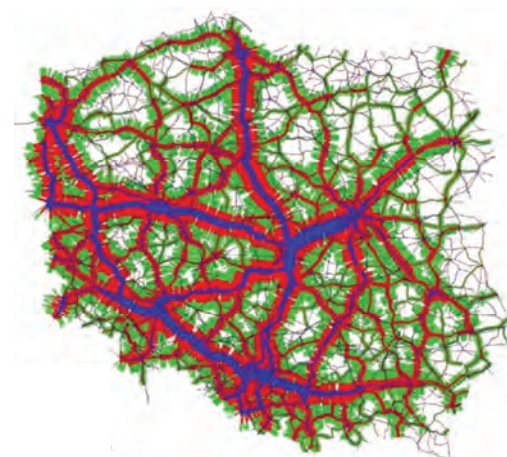


Fig. 4. The level of emissions of harmful compounds by passenger transport in 2020 in g/km (legend: blue line – HC, red line – CO, green line – NO_x) [8]

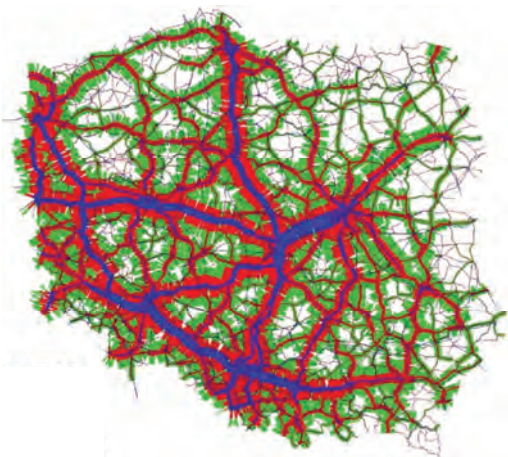


Fig. 5. The level of emissions of harmful compounds by passenger transport in 2025 in g/km (legend: blue line – HC, red line – CO, green line – NO_x) [8]

6. Conclusions

The EMITRANSYS simulation model presented in the paper is an attempt to answer the questions posed by transport decision makers that deal with proecological solutions for this system. Both the mathematical model and its implementation in the PTV VISUM program can provide tools to support decisions in the area of developing a proecological transport system at both operational and strategic levels. These decisions may concern, inter alia, modernization of the infrastructure elements of the transport system or the reorganization of traffic.

EMITRANSYS model, moreover, can be applied to:

- studies on the impact of external costs on the structure of traffic, the attractiveness of transport or between different transport service providers,
- analysis and assessment of the impact of changes in the structure of road vehicles in the field of Euro standards on the emission of harmful exhaust compounds produced by road vehicles,

- research on inter-branch interactions taking into account the costs of emission of harmful exhaust fumes emitted by means of transport,
- identification of areas in urban agglomerations in which the emission of harmful compounds exceeds the acceptable standards,
- verification of the strategy of protection of areas particularly valuable for nature in order to prevent the negative effects of transport activities,
- multifaceted assessment of greenhouse gas emissions and other harmful exhaust components related to transport activities,
- research into the strategy for the development of a sustainable transport system by analysing investments on the road and rail network, taking into account aspects of transport ecology,
- distribution of financial resources for investments and modernization of transport infrastructure elements in Poland.

Summing up thanks to the developed implementation, it is possible to simulate the assessment of the impact of decisions on the development of the transport network and legislative conditions of road transport on pollution of the environment, taking into account the functional dependence between technical and organizational consequences of these decisions and the level of harmful exhaust emissions.

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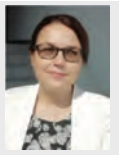
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Performance analysis of turbofan engine with additional combustion chamber fueled by alternative fuel

Presented paper is focused on the performance analysis of turbofan engine with additional combustion chamber fueled by alternative fuel. Additional combustion chamber was situated between high pressure turbine and low pressure turbine. Such configuration is also known under the name inter turbine burner (ITB). As an alternative fuels burn in additional combustion chamber were selected cryogenic fuels: liquid hydrogen and liquid methane. The main combustion chamber was supplied by conventional aviation fuel - kerosene. Thermodynamic model of turbofan engine with ITB was implemented in MATLAB environment. The problem of modeling an additional combustion chamber was presented. Analysis was carried out for take-off conditions. Obtained results indicate that implementation of ITB exerts a positive influence on engine performance resulting in thrust increase.

Key words: *alternative fuels, liquid hydrogen, liquid methane, inter turbine burner (ITB), turbofan engine*

1. Introduction

The problem associated with the emission of harmful products of combustion of hydrocarbon fuels is an inherent element of any type of transport using fossil fuels. Emissions of greenhouse gases and other harmful substances to the atmosphere, generated by the aviation sector can be reduced by increasing the efficiency of the turbine engines, improving the quality of applied fuels, modification of the combustion chamber and injector assembly, or replacing conventional aviation fuel with alternative fuel.

First of all, applied fuel cannot negatively affect engine performance, as well as, should retain its properties in a wide range of changes of atmospheric conditions (pressure and temperature changes due to the increase of altitude). The properties of aviation fuel are determined directly by the content of hydrocarbons. Turbine engines during operations must ensure high-quality atomization and stable combustion process of continuous nature. The quality of the atomization process in the combustion chamber is influenced by the physical properties of the fuel such as density, viscosity and surface tension. Alternative fuels can differ significantly in composition compared to conventional fuels, therefore it is necessary to understand the effect of the physical and chemical properties of a given fuel on combustion process [1].

Alternative fuel implementation in transport will reduce the dependency of global economy on the basic energy sources of fossil fuel. Air traffic is increasing from year to year, so the key issue is to find a sustainable alternative fuel source that will keep fuel costs at reasonable level. Another very important problem is impact on the natural environment. The aircraft transport generates about 2% of carbon dioxide (CO₂) pollutants caused by human activity, and it is estimated by Intergovernmental Panel on Climate Change (IPCC) that this figure will reach 3% in 2050 year [10].

2. Future of cryogenic fuels in aviation

Long term solution related to alternative fuels application in aircraft engines will have drastically reduce emission of greenhouse gases to the atmosphere. Due to this requirement, application can find fuels with zero carbon

content, such as liquid hydrogen (H₂) or those, with a low content of this chemical element, like liquid methane (CH₄) [2]. Application of cryogenic fuels in turbine engines, will involve modification of combustor chamber elements, fuel system, changes in the airframe configurations and huge alteration in infrastructure. It will be necessary to use a heat exchanger to evaporate the cryogenic stored fuel, before it injection and combustion [2, 3]. Necessity of application well-insulated fuel tanks of considerable weight and volume will reduce the energy efficiency of aircraft during short range flights. It is estimated that in case of short range airplanes fueled by liquid hydrogen, the decrease in energy efficiency will amount to approximately 28% during missions with a range of 926 km (500 nautical miles) [3]. According to information in the literature [3], in case of using hydrogen fuel in long range airplanes, the loss of energy efficiency will amount about 2% at a distance of about 5556 km (3000 nautical miles). It will be caused by the smaller take-off weight of the aircraft, resulting from the low mass of hydrogen fuel.

2.1. Liquid hydrogen

Hydrogen is characterized by clean combustion, high flame speed, wide burning limits, easy ignition and very good cooling capacity. Comparing properties of kerosene, hydrogen and methane, presented in Table 1, hydrogen has about 2.7 time higher energy content per unit mass than kerosene, and about 2.4 time higher than methane. To the main hydrogen disadvantages can be included low density per unit volume and low boiling point. The other problem is issue that hydrogen production is costly. Currently the most economically production of hydrogen is by steam reforming of natural gas, partial oxidation of methane or coal gasification. Electrolysis of water is another source of hydrogen but is inefficient and expensive [7].

2.2. Liquid methane

Liquid methane is the other candidate for alternative aviation fuel. Cooling capacity of liquid methane are not as large as for hydrogen, but are higher than for conventional fuel used in turbine engines. The specific energy of liquid

methane is higher as compared with kerosene (Table 1). Methane is also characterized by good thermal stability and clean combustion [7].

The disadvantage of this type of fuel will be low density and low boiling point. Methane require about 70% more volume to storage than kerosene. This may result in problems with aircraft configuration, due to considerable size of fuel tanks [7]. Because methane is also cryogenic fuel like hydrogen, both of this fuels will be accompanied by similar problems, for example such as losses of fuel due to vaporization.

The comparison of selected properties of kerosene, liquid hydrogen and liquid methane are presented in Table 1.

Table 1. Properties of selected fuels [7]

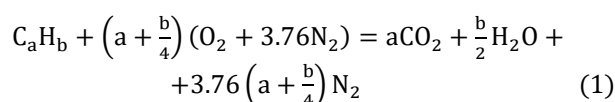
Property	Unit	Kerosene C ₁₁ H ₂₁	Liquid Hydrogen H ₂	Liquid Methane CH ₄
Lower specific energy	MJ/kg	42.8	116	49
Cooling capacity	MJ/kg	0.38–0.85	20.2	2.55
Specific heat	kJ/(kg·K)	1.97	7.32	3.43
Flame speed	m/s	0.39	2.67	0.37
Boiling point	K	423–573	21	111
Freezing point	K	223	13	91
Relative density	289–289 K	0.8	0.071	0.424

3. Combustion process

Aircraft turbine engines require high combustion efficiency at wide range of operating conditions, from sea level to 11 km of ceiling for civil aircraft. Combustion process should be free from pressure pulsation and combustion instabilities. Fuel ignition should be reliable in both, take off and cruise conditions, especially at low ambient temperature and in case of turbine engine flameout at high altitude [4].

The reaction of fuel combustion should progress with sufficient amount of oxygen. This process is connected with the definition of stoichiometric ratio. Stoichiometric ratio is the coefficient that define the complete combustion of hydrocarbon fuel which gives in results the complete conversion to CO₂ and to H₂O [4, 8].

The equation of complete combustion of general hydrocarbon fuel with air can be write as follows [4]:



The stoichiometric fuel-air ratio (FAR) can be calculated from the reaction equation (1) [4]:

$$FAR = \frac{C_aH_b}{\left(a + \frac{b}{4}\right)(O_2 + 3.76N_2)} \quad (2)$$

It is easy to notice from equation (2) that FAR depends on the fuel composition. The lowest stoichiometric fuel-air ratio has hydrogen while the highest kerosene (Table 2).

Table 2. Stoichiometric FAR for selected fuels

	Fuel		
	Kerosene	Methane	Hydrogen
FAR	0.069	0.058	0.029

The dependence of actual fuel-air ratio to stoichiometric fuel-air ratio is defined as fuel-air equivalence ratio, and is describe by equation (3) [4]:

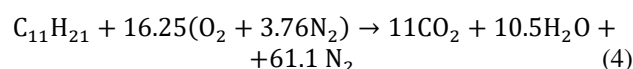
$$\phi = \frac{(Fuel/Air)_{actual}}{(Fuel/Air)_{stoichiometric}} \quad (3)$$

For stoichiometric mixture $\phi = 1$. When $\phi < 1$ the mixture is referred to lean or weak, and is characterize by excess of air. For fuel rich mixture $\phi > 1$, and due to the deficiency of oxygen, the combustion process is incomplete [4].

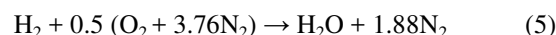
The reaction rate in the combustion chamber depends considerable on two parameters: static pressure (P) and temperature (T). At low pressure the reaction rate proceed slowly. This process may become limited at high altitudes where the pressure is lower than on the ground. The other factors that affected combustion are the rate at which the fuel is vaporized and mixed with air [4].

Product resulting from the combustion process depend on the composition of applied fuel. The chemical equation of complete combustion of selected fuels with air can be write as follows:

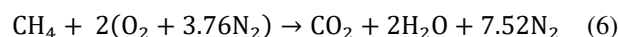
– for kerosene (with an approximate pattern C₁₁H₂₁ [7]):



– for hydrogen (H₂):



– for methane (CH₄):



The main products release during kerosene combustion are carbon monoxide (CO), carbon dioxide (CO₂), nitrogen dioxides (NO_x), water vapor (H₂O), unburned hydrocarbons (UHC), sulfur oxides (SO_x) and smoke. In contrast to hydrocarbon fuels hydrogen is named 'clean' energy carrier, due to low emission of combustion gases. During the combustion of hydrogen fuel are released water vapor, in the amount 2,5 times greater than for kerosene, and nitrogen dioxides. Methane is also characterized by clean combustion due to its low carbon content [7].

4. Turbofan aircraft engine with inter turbine burner

The concept of turbo jet engine with inter-turbine burner (ITB) was extensively studied by two researches: Sirignano and Liu [6]. Authors consider two concepts of ITB: one as continuous burning in the turbine and second as discrete interstage turbine burners [4, 6]. They carried out thermodynamics analysis of turbojet and turbofan engine showed the advantages of using ITB. The advantages of that engine configuration is improvement of engine thermal efficiency and reduction of NO_x. Researches on the possibility of improving turbojet and turbofan engines performance by using additional combustion chamber situated between

turbine stages started in the late 1990s and are still being continued [4].

In conducted research ITB is a part of conventional turbofan engine design, located between the high pressure turbine (HPT) and the low pressure turbine (LPT) (Fig. 1).

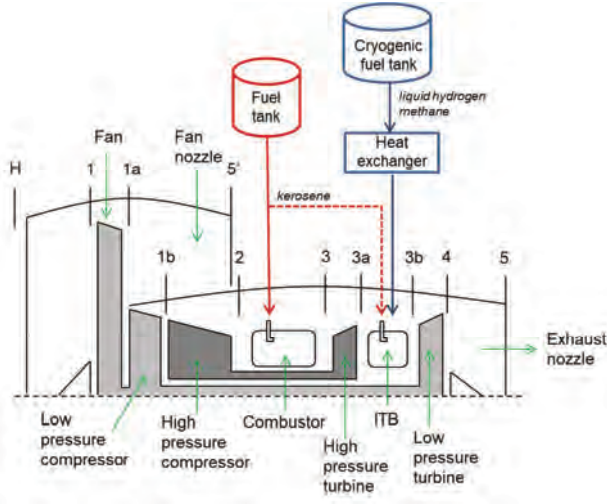


Fig. 1. Turbofan engine scheme

Actual fuel-to-air ratio relation for additional combustion chamber was calculated based on the energy balance [4].

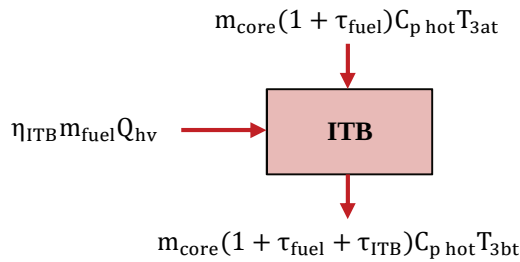


Fig. 2. Energy balance for inter turbine burner

Actual fuel-air ratio for ITB:

$$\tau_{ITB} = \frac{(T_{3bt} - T_{3at}) \cdot (1 + \tau_{fuel}) \cdot C_{p, hot}}{Q_{hv} \cdot \eta_{ITB} - C_{p, hot} \cdot T_{3bt}} \quad (7)$$

where: τ_{fuel} – actual fuel-air ratio refer to main combustion chamber, T – total temperature at given engine section (see Fig. 1), $C_{p, hot}$ – specific heat of hot gases, $C_{p, cold}$ – specific heat of air, η_{ITB} – ITB efficiency, Q_{hv} – fuel heating value.

The temperature at the outlet of the LPT was calculated from the energy balance for low pressure spool, in accordance with the following relation [4]:

$$P_F + P_{LPC} = \eta_m \cdot P_{LPT} \quad (8)$$

where: P_F – fan power, P_{LPC} – high pressure compressor power, P_{LPT} – high pressure turbine power, η_m – mechanical efficiency.

From relation (8), the temperature at the outlet of the low pressure turbine (LPT) is defined by the following equation (9):

$$T_{4t} = T_{3t} - \left[\frac{(1 + \mu) \cdot m_{core} \cdot C_{p, cold} \cdot (T_{1at} - T_{1t})}{m_{core} \cdot (1 + \tau_{fuel}) \cdot \eta_m \cdot C_{p, hot}} + \frac{m_{core} \cdot C_{p, cold} \cdot (T_{1bt} - T_{1at})}{m_{core} \cdot (1 + \tau_{fuel}) \cdot \eta_m \cdot C_{p, hot}} \right] \quad (9)$$

The performance of the jet engine describe fundamentals parameters such as specific thrust and specific fuel consumption. To calculate both of these indicators it is necessary to know engine thrust force.

Engine thrust force for turbofan engine with additional combustor chamber may be expressed by equation:

$$T = m_h \mu V'_5 + m_h (1 + f + f_{tb}) V_5 - m_h (1 + \mu) V \quad (10)$$

where: V – velocity of the flight, V_5 – velocity of exhaust gases (hot nozzle), V'_5 – velocity of bypass air (cold nozzle), μ – bypass ratio, m_h – mass flow rate passing through the main engine core, f – fuel to air ratio, f_{tb} – fuel to air ratio refer to turbine burner.

Specific thrust is defined as a relation between engine thrust and mass flow rate:

$$ST = \frac{T}{m_h} \quad (11)$$

Specific fuel consumption is defined as the fuel flow rate per unit thrust and can be expressed by following equation:

$$SFC = \frac{m_{fuel}}{T} \quad (12)$$

where: m_{fuel} – total fuel burn in both combustor chambers, T – engine thrust.

5. Scope of the work

The main scope of presented paper is performance analysis of turbofan engine with additional combustor chamber fueled by alternative fuel. Main combustor chamber is supplied by conventional aircraft fuel. As an alternative fuels were chose cryogenic fuels such as liquid hydrogen and liquid methane. The thermodynamics model of turbofan engine with dual combustor chamber was implemented in Matlab software. Engine performance analysis was carried out for take-off conditions (off-design) as the most critical stage of the flight. The working gas was modeled using a semi-perfect gas model. Properties of the exhaust gases were modeled based on the reference [5]. The turbine cooling process was not taken into consideration. Turbine inlet temperature (TIT) for high pressure turbine was assumed as $T_3 = 1500$ K, whereas for low pressure turbine as $T_{3b} = 1300$ K. Configuration of turbofan engine with ITB and marked characteristic engine cross-sections is presented on Fig. 1. The thermodynamic model was elaborate based on the equation describing the work of main engine components, based on the reference [4, 5, 9].

Input parameters necessary for engine performance analysis are presented in Table 3.

Table 3. Input parameters for engine performance analysis

Property	Description	Unit	Take-off
Altitude	H	m	0
Velocity of flight	Ma	–	0
Air mass flow	m	kg/s	670
Fan pressure ratio	π_F	–	1.65
Bypass ratio	μ	–	4.4
Low pressure compressor ratio	π_{LPC}	–	1.6
High pressure compressor ratio	π_{HPC}	–	12.8
Overall pressure ratio	π	–	33.8
HPT turbine inlet temperature	T_3	K	1500
LPT turbine inlet temperature	T_{3b}	K	1300

Additional combustor chamber will use the air with large content of combustion gases, similar like afterburner [4].

The stoichiometric equation allow to calculate the amount of air necessary to burn given amount of fuel. Obtained results indicate that implementation of hydrogen will reduce the required air necessary to complete combustion and will allow for reduction of engine diameter. This will result in lower engine mass and lower drag force caused by the engine overall dimensions and increase the range of the flight.

6. Result discussion

Results of conducted analysis are presented in Table 4 and Table 5. All fuels applied in additional combustor chamber represent lean-fuel mixture (Table 2). For take-off conditions, the highest value of specific thrust (ST) is obtained for turbofan engine with ITB supplied by hydrogen. The lowest value is exceeded for kerosene fuel, which is caused by the differences in heat value of each applied fuel. Due to large heat value of hydrogen fuel (Table 1), the fuel-to-air ratio is lower than for other selected fuels, which implies lower fuel consumption by the engine.

Table 4. Results of thermodynamics analysis of modified turbofan engine configuration

			Turbofan engine with ITB		
			Kerosene	LH ₂	Methane
Specific thrust	kj	[Ns/kg]	392.19	423.77	409.13
Specific fuel consumption	cj	[kg/Ns]	$1.16 \cdot 10^{-5}$	$9.35 \cdot 10^{-6}$	$1.09 \cdot 10^{-5}$
Engine thrust	K	[kN]	262.77	283.93	274.12
Fuel-air-ratio	τ_{fuel}	–	0.0195		
Fuel-air-ratio	τ_{ITB}	–	0.0050	0.0018	0.0044
Fuel mass flow	m_{fuel}	[kg/s]	2.43		
Fuel mass flow	$m_{fuel, ITB}$	[kg/s]	0.63	0.23	0.55
Exhaust mass flow	m_{hot}	[kg/s]	127.13	126.73	127.05
Velocity of exhaust gases	c_5	m/s	734.57	903.85	824.38
Velocity of cold gases	$c_{5'}$	m/s	310.27		

Additional analysis was carried out, to compare the effect of the additional combustion chamber on the performance of conventional engine design. The results are presented in Table 5.

Application of additional combustion chamber affects the increase in thrust value generated by the turbine engine. Comparing the classical configuration of turbofan engine with modified configuration with ITB supplied by kerosene, the engine specific thrust was increased by 4.2%. For the case of supplying the additional combustion chamber by hydrogen, engine specific thrust increased by 12.6% and by methane by 8.7%. Interesting is the fact that the specific fuel consumption (SFC) for modified engine is slightly lower than for conventional engine design. SFC is defined as the fuel flow rate per unit thrust. For the engine with additional combustion chamber, fuel flow rate is higher by the value of fuel mass flow in additional combustor. Engine thrust express by a formula (7) depends on the velocity of exhaust gases, which is about 38% higher for engine with dual combustor system than for base design. In addition obtained engine thrust is higher so in accordance with relation (9), the SFC must decreased. SFC is very important factor that describe engine economy.

Table 5. Results of thermodynamics analysis of modified turbofan engine configuration

			Turbofan engine
			Kerosene
Specific thrust	kj	Ns/kg	376.39
Specific fuel consumption	cj	kg/Ns	$9.62 \cdot 10^{-6}$
Engine thrust	K	kN	252.18
Fuel-air-ratio	τ_{fuel}	–	0.0195
Fuel mass flow	m_{fuel}	kg/s	2.43
Exhaust mass flow	m_{hot}	kg/s	126.50
Velocity of exhaust gases	c_5	m/s	654.51
Velocity of cold gases	$c_{5'}$	m/s	310.27

The relationship between engine unit parameters and engine overall pressure ratio is presented on Fig. 3, for base and modified engine. For classical configuration of turbofan engine, the maximum of ST is exceed for lower values of engine pressure ratio than for configuration with additional source of heat. In addition after reaching the maximum, the value of ST decrease violently in comparison with the graphs that describe the changes of ST with engine pressure ratio for modified engine configuration.

The highest values of ST with simultaneous the lowest values of SFC are obtained for engine configuration with dual combustor chamber, supplied by the hydrogen.

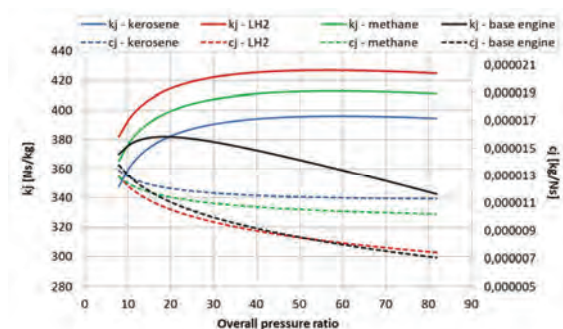


Fig. 3. Specific thrust and specific fuel consumption in function of engine pressure ratio

The relationship between engine unit parameters and LPT inlet temperature is presented on Fig. 4. In general, the ST is increase with the inlet turbine temperature increase, which is accompanied by the growth of specific fuel consumption. It is noticeable that the growth of SFC in case of application hydrogen fuel is lower than for kerosene and methane at given TIT. The highest values of specific thrust with simultaneous the lowest value of specific fuel consumption are obtained for hydrogen, which is caused by its properties. Even for higher temperature values, the SFC for hydrogen is much smaller than for kerosene and methane.

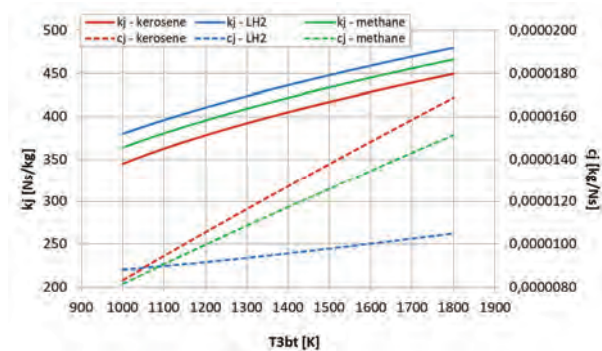


Fig. 4. Specific thrust and specific fuel consumption in function of LPT inlet temperature

Nomenclature

HPT high pressure turbine
ITB inter turbine burner
LPT low pressure turbine

SFC specific fuel consumption
ST specific thrust
TIT turbine inlet temperature

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Selection of ion sensor operating parameters in quasi-static conditions

Evaluating combustion quality using sensors that allow continuous assessment of the process is one of the modern methods of engine sensory diagnostics. The proper calibration of such systems is a task that requires many studies to determine the conditions and quantities affecting the process. The analysis of significance of quantities related to the ionization signal was carried out in the article. The magnitude of the voltage generating the electric field, the type of spark plug used, the distance of the spark plug electrodes and the dynamic factor – spark plug operating temperature – were all tested. The tests were carried out using a CNG burner (with an excess air ratio of $\lambda = 1$) and four spark plugs. As a result of the ionization signal research, the following relationships were obtained: regarding the impact of the sensor position on the amount of generated voltage (the smaller the distance the greater the value of the signal), the effect of temperature on the sensor resistance (non-linear relationship: increase in temperature decreases resistance, with $R^2 = 0.9997$) effect of system voltage on the ionization signal (linear relationship: voltage increase increases the ionization current signal with a determination coefficient of $R^2 = 0.9803$). In addition, it was found that using an iridium electrode candle had the best effects on the ionization current, regardless of the electrode's geometrical parameters.

Key words: ionization sensor, engine control, combustion process diagnostics

1. Introduction

The ionization signal is used in many engine applications. Popular brands such as Mazda, Saab, BMW and Harley-Davidson [4] successfully use knock detection systems based on the ionization signal. The efficiency and fast response time of such a system has also been successfully used in Formula 1 engines [9]. There are also other, less popular solutions, that use the phenomenon of ion formation in the cylinder to measure EGR content [8], determine the quality indicators of the combustion process [2] or control the operation of the HCCI engine [7].

The nature and amplitude of the ionization signal is a derivative of many factors, both static and dynamic, i.e. variable over time. Determining their effect on the signal allows the use of the ionization sensor as an advanced diagnostic tool in modern internal combustion engines, and thus allows to improve the engine's operating indicators and reduce harmful exhaust emissions.

One of the most important factors varying in time is the density of ions and free electrons between the spark plug electrodes that allow a free flow of current. The more ions and free electrons are available, the more ionization current could potentially flow in the circuit. The presence of ions in an internal combustion engine results mainly from the phenomenon of chemical and thermal ionization of gas [4]. Both phenomena strongly depend on the process temperature [6, 11]. The degree of gas ionization by radiation depends on the temperature as described by the Saha's formula [5]:

$$\frac{N_j N_e}{N_n} = 2 \frac{u_1}{u_0} \frac{(2\pi m k T)^{3/2}}{h^3} e^{-E_0/kT} \quad (1)$$

where: N_j – ion concentration, N_e – free electron concentration, N_n – non-ionized atoms concentration, m – electron mass, k – Boltzmann's constant, h – Planck's constant, T – gas temperature, E_0 – ionization energy, u_0 and u_1 – so-called partition functions specific to a given chemical element.

Another factor affecting ion density is the type of fuel used, as well as its octane number [10]. Eriksson [3] indicates that the so-called chemical phase curve rise is related to the composition of the air-fuel mixture.

The current flowing through the sensor also depends on the design of the electrical system. While the resistance of the sensor itself can be classified as variable over time (due to the change in sensor temperature, and thus its material resistance), the other elements of the measuring system can be characterized as static.

The characteristics of the sensor itself, i.e. the spark plug, can be named as one of the static factors affecting the ionization signal. The structure of the electrodes, their number, materials used and even the heat value of the spark plug [1] have a direct impact on the signal amplitude. Similarly, the electrical parameters of the system, such as the intensity of the electric field generated by the high voltage system, as well as the measuring resistance itself contribute to the current flow parameter [3, 5].

There are a number of factors conditioning the formation of the ionization signal: the presence of soot on the spark plug electrodes, air humidity, the rate of ion formation [3], and others.

The aim of this article is to try to select the ionization sensor operating parameters in conditions close to static, so that it is possible to determine the interacting factors and obtain the parameters of a universal measuring system that allows repeatable measurements of the ionization signal and their unambiguous interpretation in the future.

2. Method

2.1. Test stand

A Teclu laboratory gas burner was used to achieve measurement conditions similar to static (Fig. 1). The fuel burned in the burner provided a stable, constant source of

ions and free electrons generated in the flame as a result of the thermal and chemical ionization process.

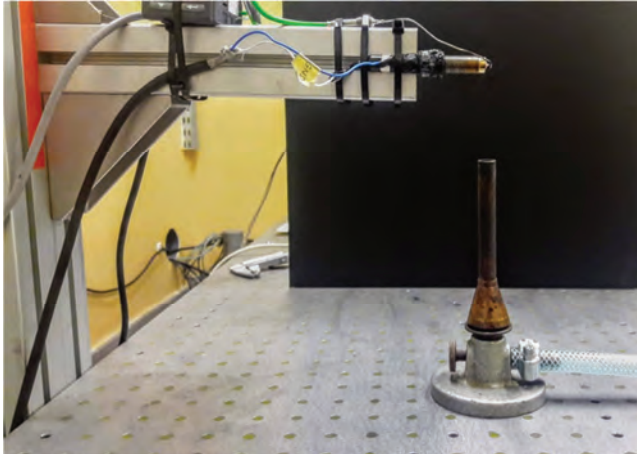


Fig. 1. Teclu gas burner test stand

The measuring element used was the spark plug typically used in spark-ignition internal combustion engines. The spark plug was attached to a structure that allowed changing its height. A K-type thermocouple with a maximum measuring range of 1100°C was used to determine the operating conditions of the ionization sensor, whose tip with a diameter of 1 mm was located near the measuring electrodes (Fig. 2).

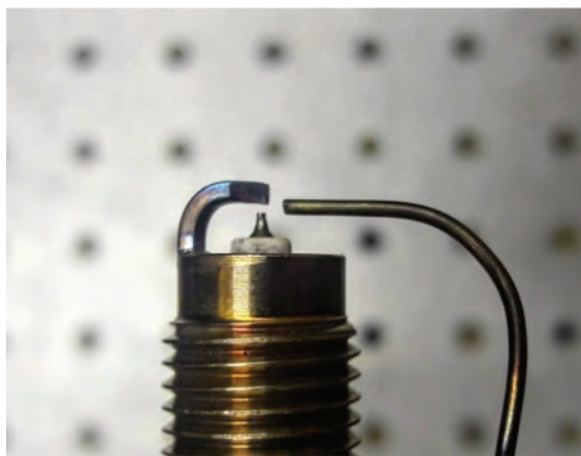


Fig. 2. Measurement of the ionization sensor operating temperature

The amount of gas supplied was determined using a Bronkhorst 111B flow meter. In order to determine the air excess coefficient λ , the burner settings were calibrated using an exhaust gas oxygen sensor – Bosch LSU 4.9.

Diagram of the ionization current measuring circuit is shown in Figure 3. The source of the electric field was a DC power supply with an adjustable voltage range of 50–388 V. The direct measurement element was a resistor with a variable resistance setting in the range of 0–1.14 M Ω . According to Ohm's law, the ionization current flowing in the circuit generated a proportional difference in potential across the resistor. The ionization voltage thus obtained along with the other signals (temperature, gas mass flow) was recorded by the AVL IndiCom data acquisition system.

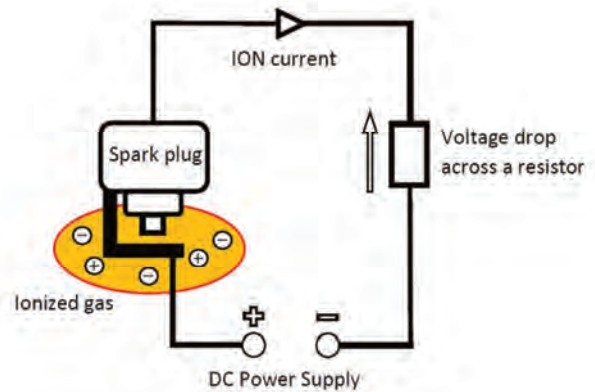


Fig. 3. Electric schematic of the measurement system [4]

2.2. Measurement points

The adopted research strategy was aimed at determining the impact of the main static factors on the signal, so that in the future it is possible to clearly interpret the results regardless of the research conditions.

The main static factors were determined in the article as:

- voltage generating an electric field,
- spark plug type used,
- spark plug electrode distance.

As well as the dynamic factor:

- the operating temperature of the spark plug affecting the resistance of the system.

The parameters of the burner and the measuring system used to determine the influence of static factors are presented in Table 1. The characteristics of the spark plugs used are shown in Table 2.

Table 1. Parameters of the burner and the ionization sensor

λ	1	1	1
Q_CNG [g/h]	31.6	31.6	31.6
Spark plug number	1	1–4	2
System voltage [V]	50–388	200	200
Resistance [M Ω]	1.142	1.143	1.143
Measurement height [mm]	8	8	8

Table 2. Parameters of the spark plugs used

Spark plug designation	NGK T20379G	Denso XE20HR-U9	Denso X27ETR	Denso IXEH22TT
Spark plug number	1	2	3	4
Resistance [k Ω]	1.65	4.72	6.33	5.71
Electrodes separation [mm]	0.45	0.4; 0.8; 1.2	0.7x2	0.95
Electrode diameter [mm]	0.7	2.1	2.2	0.5
Electrode material	iridium	nickel	nickel	iridium

3. Results

3.1. Impact of sensor location on the amount of generated voltage

Various sensor positions relative to the burner ranging from 0.8 to 11 cm distance were analyzed. The criterion for choosing the measuring point was a stable and high concentration of ions and free electrons in the flame (Fig. 4). Ini-

tial, experimental recognition allowed the determination of the optimal measuring point (position 0.8 cm above the burner), which was used in subsequent research. It was observed that as the distance from the burner increased, an ionization signal with a lower value and less stability was obtained.

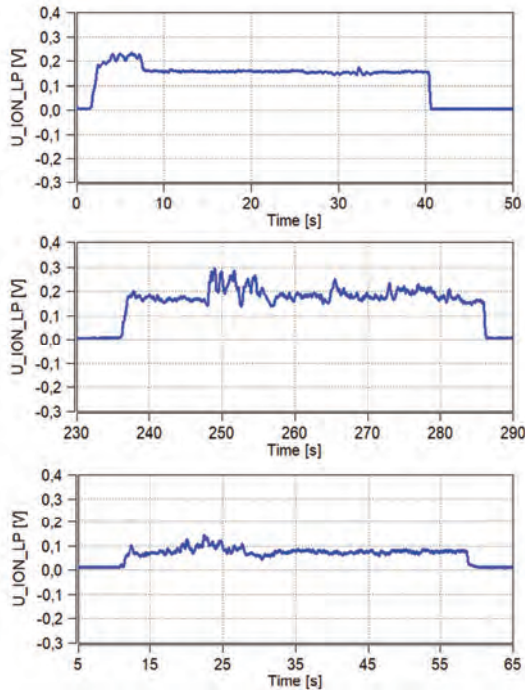


Fig. 4. Ionization voltage signal waveform; measuring points, from the top: 1, 3 and 8 cm above the burner (spark plug No. 2)

3.2. Impact of temperature on the sensor resistance

As a result of the combustion process in the cylinder, the temperature of the engine components increases, which includes the spark plug, which also functions as an ionization current sensor. As a result of this phenomenon, the spark plug resistance values in the temperature range 0–175°C were determined in order to determine the possible influence of the sensor temperature on the size and nature of the ionization current signal. An increase in the resistance value of one of the elements causes a change in the electrical parameters of the entire system. Figure 5 shows the change in resistance of spark plug number 1 as a function of its temperature. Based on the relationship $R = f(T)$, the regression equation was determined, describing the value of resistance R as a function of the spark plug temperature T :

$$R = 2 \cdot 10^{-5}T^2 - 0.0078T + 1.8544 \quad (2)$$

Changing the temperature from 0 to 175°C causes the resistance of spark plug number 1 to decrease by 43%. Assuming that the average value of current I flowing during engine operation is about 6 μA (typical value for an SI engine powered with natural gas) and the voltage value U of the system is about 300 V (typical value for ionization sensors), then as a result of Ohm's law the theoretical value of system resistance can be obtained:

$$R = \frac{U}{I} = \frac{300 \text{ V}}{6 \mu\text{A}} = 50 \text{ M}\Omega \quad (3)$$

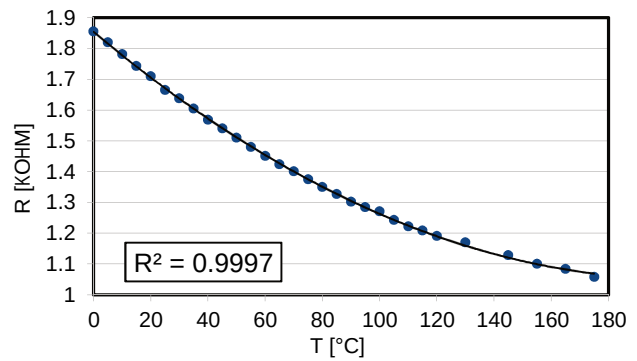


Fig. 5. Impact of sensor temperature on the value of its resistance (spark plug No. 1)

So (in this particular case), a change in the resistance of one of the circuit components, which is the spark plug (No. 1) by 43%, causes a change in the resistance of a typical measuring system by 0.0016. Thus the effect of the sensor temperature on the ionization signal was concluded to be small, or even negligible; however, this applies to the circuit used by the authors.

3.3. Impact of system voltage on the ionization signal

According to previous studies, the sensor was centered in the flame at a height of 8 mm above the burner. The voltage value was increased in increments of 50 V until it reached its maximum (Fig. 6). Figure 7 shows the measurement results of the ionization current relative to the system voltage.

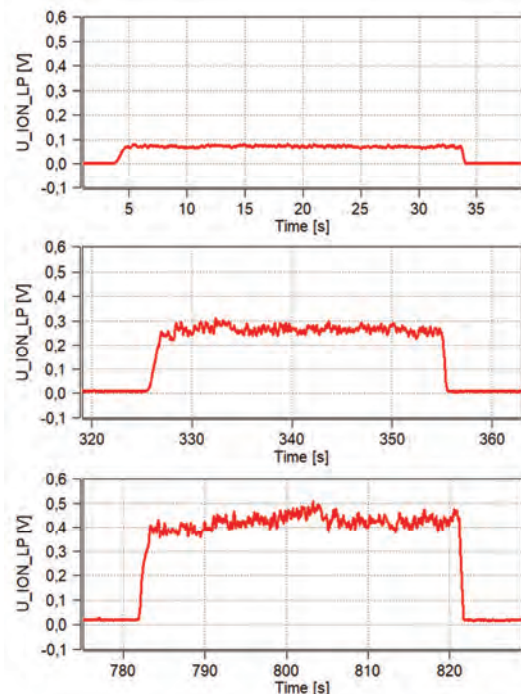


Fig. 6. Ionization waveforms with system voltage of (from the top): 50 V, 200 V and 388 V (spark plug No. 1)

As a result of the voltage increase, an increase in the ionization current was observed, which is directly related to the increase in the electric field intensity between the spark plug electrodes. In addition, it is potentially possible to

further increase the value of the flowing ionization current by increasing the system voltage. Thus the so-called saturation current can be reached, after which all the ions formed around the spark plug electrodes will be captured by the sensor.

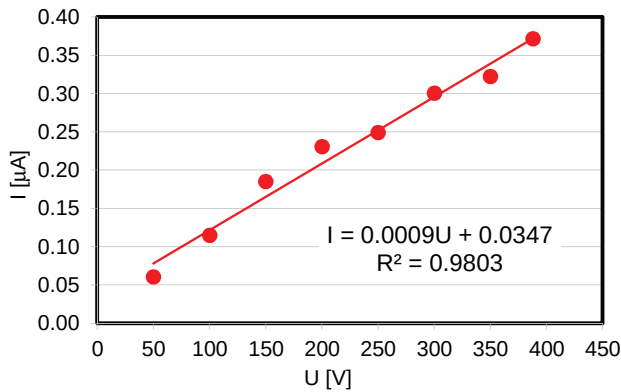


Fig. 7. Impact of the measuring system voltage on the ionization current value (spark plug No. 1)

3.4. Impact of the selected spark plug type

In order to determine their effect on the ionization current signal the characteristics of four types of spark plugs were presented in Table 2. In addition, the influence of the electrodes separation distance on the received signal was examined for the spark plug number 2. These distances were: 0.4, 0.8 and 1.2 mm. Each of the spark plugs was located centrally in the flame of the burner at a height of 8 mm. The value of the system voltage was 200 V, and the excess air coefficient λ was set to a value of 1. The waveforms of the (selected) measured signals are shown in Fig. 8 and the collective results in Table 3.

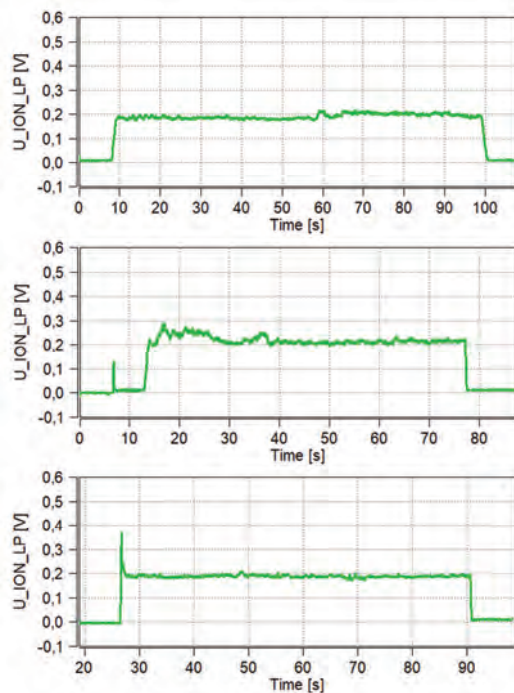


Fig. 8. Ionization signal waveforms using spark plug 2 with equal spacing (from the top): 0.4, 0.8 and 1.2 mm

Table 3. Ionization signal measurement results for selected spark plugs

Spark plug number	Electrodes separation [mm]	Mean ionization voltage [V]	Mean ionization current [μ A]
1	0.45	0.221	0.193
2	0.4	0.201	0.176
2	0.8	0.214	0.187
2	1.2	0.186	0.163
3	0.7 x 2	0.206	0.180
4	0.95	0.222	0.194

The average ionization current values indicate that spark plug 4 has the highest capacity to attract ions and electrons from the flame, and the smallest spark plug 2 with the electrode separation distance equal to 1.2 mm. The sensor No. 4 electrodes are made of iridium, are 0.95 mm apart and the diameter of the central electrode is 0.5 mm. Spark plug 2 has nickel electrodes, the distance between them is 1.2 mm, and the diameter of the positive electrode is 2.1 mm.

The experiment has also shown that the position of the sensor relative to the flame is important in relation to the ionization current signal received. Spark plug No. 2 was located in the center of the flame, but during the second test the current value was 3% higher than in the first. In order to accurately identify the impact of the spark plug used, it is necessary to repeat the tests using an ionization sensor located in the internal combustion engine or to increase the precision of the spark plug positioning in the flame of the burner.

4. Conclusions

Based on the research and analysis, a large impact on the ionization current was found for the spark plug separation distance, sensor temperature, system voltage and type of spark plug used.

The following detailed conclusions have been drawn:

- a change in the distance between the spark plug and the energy source (burner) significantly changes the generated ionization current values. The optimal value was set at 8 mm;
- the operating temperature of the spark plug has the opposite effect (but not linearly) on its resistance. Resulting in the relation: $R = 2 \cdot 10^{-5} \cdot T^2 - 0.0078 \cdot T + 1.8544$, with the coefficient of determination equal to $R^2 = 0.9997$;
- the relation between the system voltage and the ionization signal is proportional to the value of this voltage: this can be denoted as: $I = -9 \cdot 10^{-7} \cdot U^2 + 0.0013 \cdot U + 0.0035$ (where $R^2 = 0.9803$), which means that every voltage change of 50 V changes the ionization current by about 0.045 A;
- a spark plug with an iridium electrode provided the most favorable ionization current results, independently from the distance between the electrodes.

The conducted tests confirmed the need for further research related to the ionization signal, which may soon become a diagnostic signal allowing the assessment of subsequent aspects of the combustion process (development of the flame front, its propagation speed or other parameters).

Nomenclature

I	ionization current	R	resistance
U	voltage	λ	air excess ratio
N	concentration		

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Stress analysis of the cylinder block of a small compression ignition engine

The work contains calculations to determine the deformation and stress in the block of a currently produced small displacement compression ignition engine. It is also an attempt to introduce some modifications to reduce the mass of the calculated component. In the first step, based on measurements, the model of the engine block was developed. The Autodesk Inventor 2016 software was used. Two additional components were also designed to provide the block closure: a simplified cylinder head and an integrated main bearing support. All elements were imported to the Siemens NX 12 program. The calculations were carried out for different cylinders and different values of the combustion pressure. An attempt was made to introduce some modifications to reduce the weight of the calculated element.

Key words: Diesel engine, cylinder block, modelling, finite element method

1. Introduction

Nowadays we can observe intensive development of vehicles powered by electric motors. Such motors are fed with electricity both by energy stored in batteries charged externally and generated on the fly in the case of hybrid drivetrain units. The latter ones, in majority of the cases, make use of spark ignition combustion engines. However, American auto makers successfully use compression ignition engines (CI) in hybrid systems as power units for luxury cars. European automotive OEMs have introduced the CI engines to powering such type vehicles as well. These are engines of small displacement, which are also mounted in cars as a conventional source of power. In a passenger car, the engine operated at variable speeds and loads has to be lightweight, cheap in operation, and should provide a good power band. Due to the above factors, reduction of mass of engine components becomes a necessity. Components of crank and piston system have to be lightweight and because of frequently changing velocity should feature low moment of inertia; additionally forces which act on these elements frequently change their direction. The cylinder block, however, is the heaviest element of the engine. The cylinder block should be as stiff as possible to assure durability of the power unit. Methods of calculating the cylinder block which are used today enable full usage of available material. Implementation of the Finite Element Methods to the calculations has allowed considerable weight reduction of manufactured power units.

The cylinder block of a combustion engine is a component which integrates all of the mechanisms of the engine. Because of this, a lot of care has to be given in the process of designing this element, focusing mainly on its stiffness and mass, stiffness being the more important aspect of the two. A cylinder block serves as a support for main bearings of the crankshaft, enables kinematic positioning of pistons, as well as provides the mounting points for cooling and lubrication systems. The cylinder block also contains engine timing, chassis and gearbox mounting points. Floppy cylinder block can deform during engine operation – under changing and high loads – which in turn decreases the durability and reliability of the engine [6, 7].

Cylinder blocks of piston engines are manufactured from cast grey iron or aluminum-silicon alloys. Cylinder

blocks for automotive engines of small swept capacity in majority of the cases are made of grey iron, cast as one element along with the cylinders. In such case a better grade of grey iron, with higher abrasion resistance should be used. An integrated main bearing support made out of cast aluminum-silicon alloy is then affixed to the engine block from underneath, usually along with a stamped steel oil pan screwed in. Such solution ensures required stiffness, optimal parameters in regards to noise, vibrations, harshness (NVH), and simplified manufacturing processes within limits of assumed dimensional limitations [2]. Manufacturing costs of a cylinder block are equal to about 10% of total engine costs, while costs of the material amounts to half of this value. Nowadays more methods of reducing manufacturing costs have surfaced, including modular engine design – e.g. designing an engine family consisting of 3- and 4-cylinder engines, which share pistons and conrods, and only slightly differ in dimensions of the crankshaft, cylinder block and cylinder head [1].

The study is devoted to calculations aimed at evaluation of strain and stress levels in cylinder block of a currently manufactured compression ignition engine of small swept capacity. It is also an attempt to introduce some modifications to reduce the mass of the calculated component.

2. Object of the investigation and its modelling

The analyses were performed for the cylinder block having the main technical parameters as specified in the Table 1.

Table 1. Main technical parameters of the engine

Engine	Compression ignition, turbocharged, common rail system	
Swept capacity	dm ³	1.248
Maximum power	kW/rpm	55.2/4000
Maximum torque	Nm/rpm	190/1500

In the first stage of the work, the cylinder block has been prepared as a 3D model to be used later on in as a math model. The Autodesk Inventor 2016 program was used for this purpose. Great effort has been made to reproduce the most important dimensions of the cylinder block as accurately as possible. Selected view and cross-section are presented in the Fig. 1 and 2.

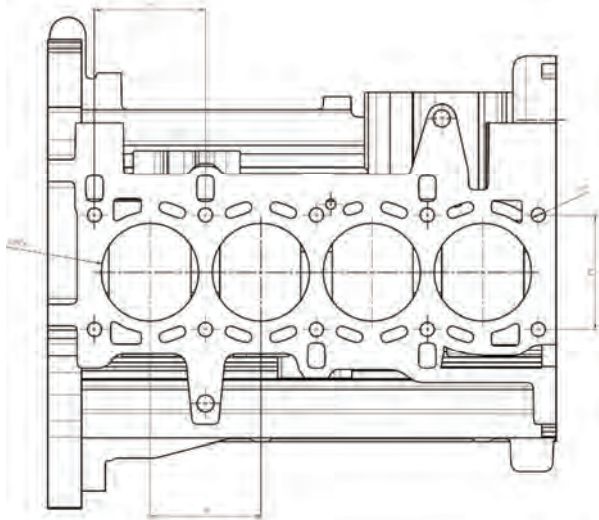


Fig. 1. Top view of the cylinder block

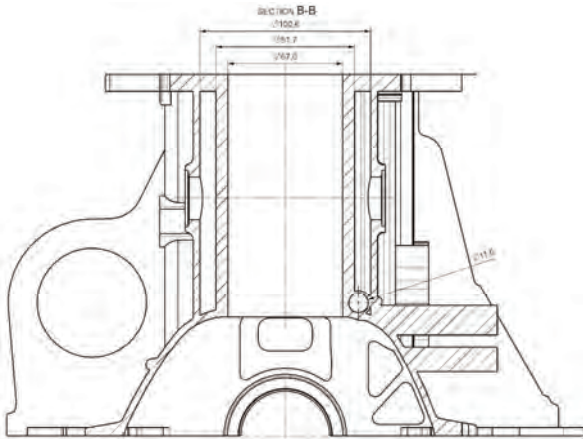


Fig. 2. Cross-section through the second cylinder's axis

Two additional elements have been modelled in the next stage, used as the closure of the cylinder block: simplified cylinder head and integrated main bearing support. Such elements were introduced to allow calculations of the screw connections between those components.

Prior to start of the FEM calculations it is necessary to prepare model suitable for such type calculations. Therefore, the model has to be simplified as much as possible without affecting the overall stiffness of the structure. It is a frequently used solution to shorten the calculation time. Features such as chamfers, small fillets and holes were removed. In case of the analyzed object most of these elements have been eliminated, however, oil channels and mounting points from both sides of the block, used to attach various engine components were not removed. The model prepared in such way was then imported to Siemens NX 12 software. Cylinder head and an integrated main bearing support were also modeled in this program for calculation purposes. View of the model prepared to the calculations is shown in the Fig. 3.

In the next step the size of the mesh was determined. In case of FEM calculations, using 3D elements as finite elements, the size of the elements has a very significant effect on duration of the calculations and their accuracy. The

cylinder block, however, is a rather big element. In the end it has been decided to use an overall mesh size of 6 mm, but using localized mesh density of 1mm, particularly around the screw mounting points. The mesh with the Jacobian matrix determinant of 10 was used.

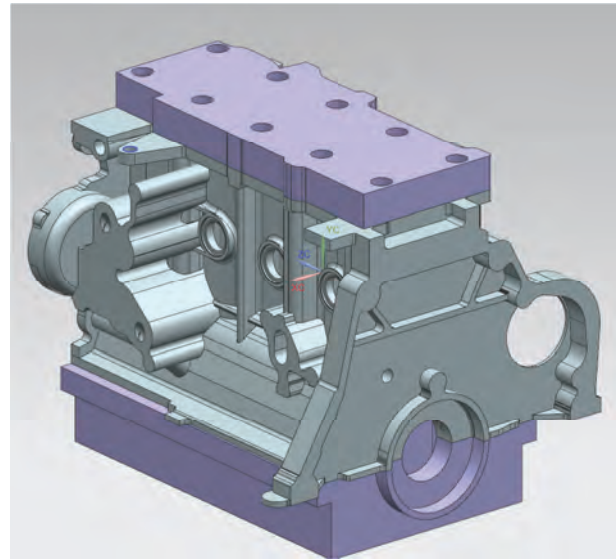


Fig. 3. Isometric view of the calculation model

3. Properties of a CI engine cylinder block mathematical model

The following constraints were placed upon the calculation phase:

- mesh elements of tetrahedral (10) type – triangle pyramids described by 10 nodes (vertices and central points of sides); mesh density 6 mm,
- mesh density locally scaled down to 1 mm, particularly around points where the loads were applied. Such step was necessary in order to shorten the duration of the calculations – in this configuration the FEM model took up to 4 hours to calculate,
- fixed constraints on elements of the cylinder block used to attach the engine – from front (Fig. 4) and rear side of the cylinder block,

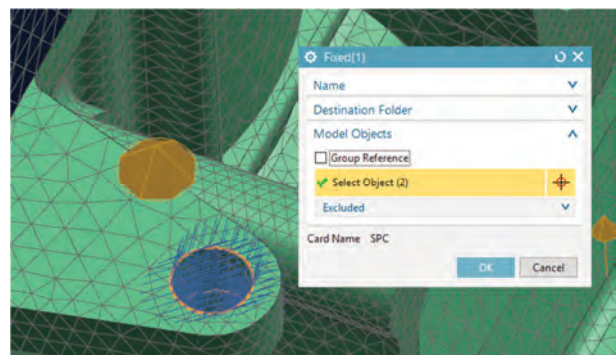


Fig. 4. Isometric projection of the calculation model

- contact between surfaces: cylinder block-cylinder head and cylinder block – integrated main bearing cover with friction coefficient of 0.3 (Fig. 5),

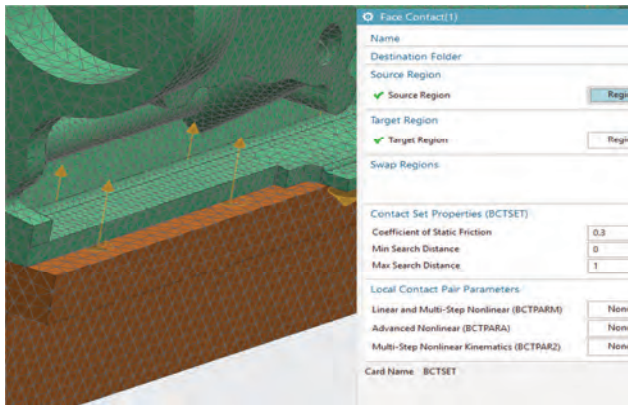


Fig. 5. Mating surface of the cylinder block with the main bearing cover

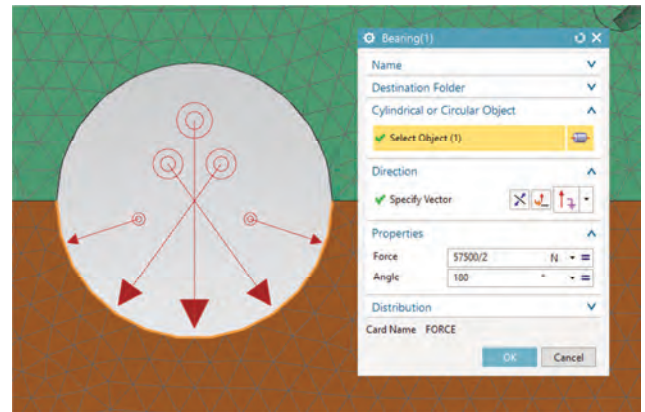


Fig. 8 Main bearing force load

- maximum combustion pressure exerted on the surface of the cylinder head at 15 MPa (Fig. 6),

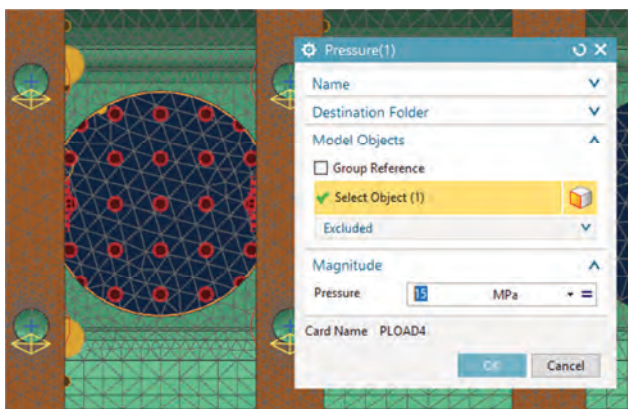


Fig. 6. Combustion pressure load placed upon the cylinder head – bottom view, along the cylinder axis

- cylinder head bolt pre-load of 22,000 N (Fig. 7),

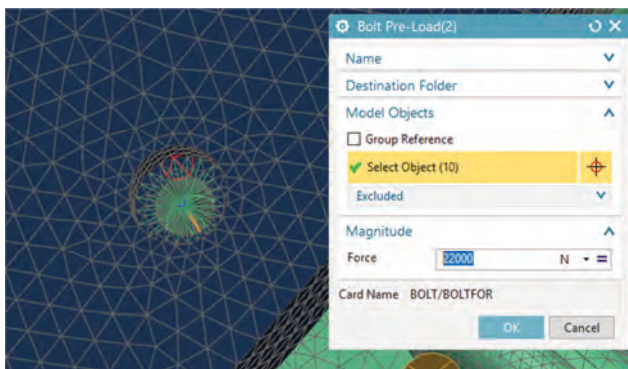


Fig. 7. Cylinder head bolt pre-load

- integrated main bearing support bolt pre-load at 12,000 N,
- cylinder 1 main bearings laden with force of 28,750 N, equal to half the maximum gas force in the combustion chamber (Fig. 8),
- the presence of washers simulated in the bolt connection model (Fig. 9) has also been introduced. Such operation was done to reduce the number of points with local stress spikes far exceeding the expected values.

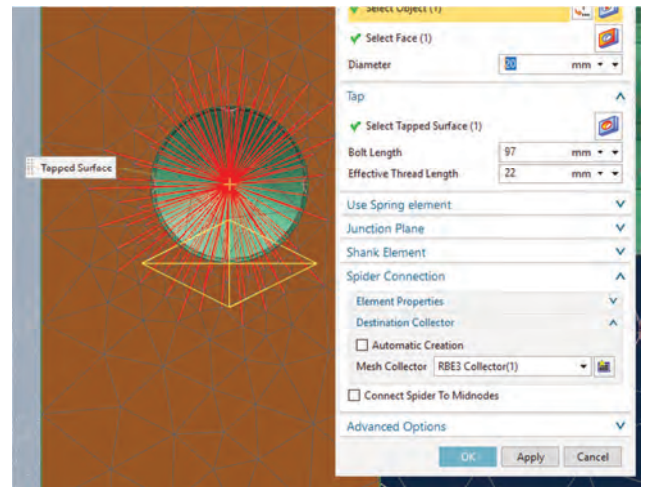


Fig. 9. Bolt connection model at the crankshaft main bearing support

4. Results of the calculations

Analysis of obtained results was performed after completion of the calculations. Maximum deformations have been spotted in the tightening area of the two central screws of the integrated main bearing support – they amount to 0.083 mm (Fig. 10).

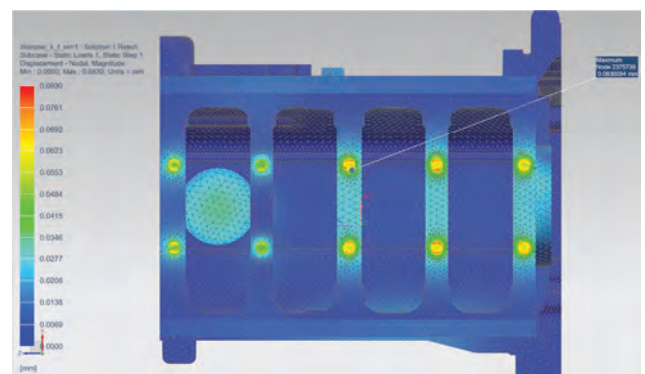


Fig. 10. Deformations of screw mounting points in the lower part of the cylinder block

In turn, the supporting points of the crankshaft in the closest location to the loaded cylinder were deformed with 0.007 mm. Small deformations of the cylinder under the effect of exerted loads have also been noticed. These deformations amounted to 0.001 mm (Fig. 11). High stiffness

in this area ensures durability of the engine and proper seal of the combustion chamber, which in turn assures a correct combustion cycle process and longevity of the engine. The smaller deformations of the cylinder, the lower risk of damage to the crankshaft-piston system, and the lower the risk of blow-by from the combustion chamber to the crankcase.

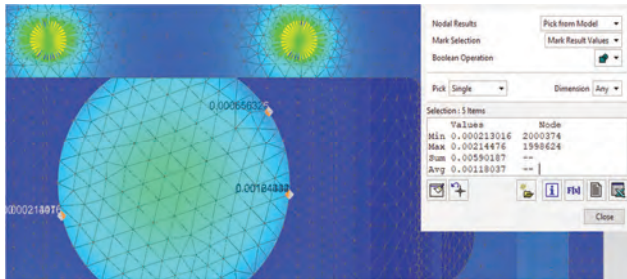


Fig. 11. Deformation of engine's cylinder

The deformations in the area of the connection between the cylinder head and the cylinder block are not considerable, their maximum value amounts to 0.01 mm.

Reduced von Mises stresses were also analyzed. The support points of the crankshaft are not heavily stressed – the mean stresses are equal to 29 MPa, while the maximum ones amount to 44.93 MPa. However, in the nodes where the screws of the integrated main bearing cover were tightened, occurrences of considerable stress spikes has been noticed. Further analysis of the stress values in these nodes it can be noticed that the smaller individual elements of the model, the higher the stress. This is the problem caused by so called FEM Singularity (Fig. 12).

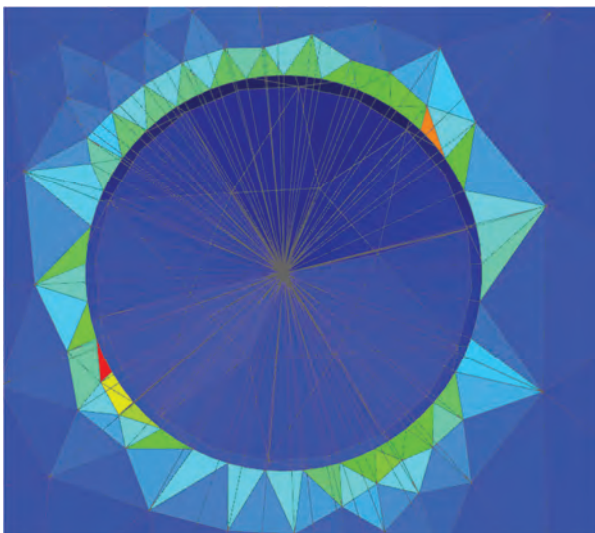


Fig. 12. Stress of the screw connection of the third mounting point of the integrated main bearing support

Such singularities usually occur in areas of sharp breakdown of edges and point-type loads. This can be caused by the method of modelling of the screw connections. In case of the analyzed calculation model, the Bolt Connection function in the NX program was used; this function connects finite element nodes laying on predefined diameter around the axis of the screw connection, to which the force was applied. Thus, it isn't applied as uniform pressure, but

rather as a series of point loads [3–5]. In the literature we can often find statements about possibility of neglecting the stress spikes present in these nodes [3, 4]. In the Fig. 12 the highest obtained nodal values of the stresses are equal to 1182 MPa. However, in the adjacent elements the stresses amount to 220 MPa. Bigger elements connected with edge of the hole feature stress levels amounting at average to 180 MPa. It should be remembered, however, that the performed simulation assumes using screws with head flanges. The washers themselves were not taken into consideration, which in this case would take over a part of the stress.

The thread in the tightening area of the screws of the cylinder head is subjected to stress of 115 MPa at the average. Lower stresses – at the average 51 MPa – have been observed on the threads of the screws the integrated main bearing cover. Axial stress in the screws of the cylinder head of the first cylinder are equal to 76 MPa at the average (Fig. 13). The screws of the other cylinders are a little bit less stressed; in these areas the stress amounts to 69 MPa at the average.

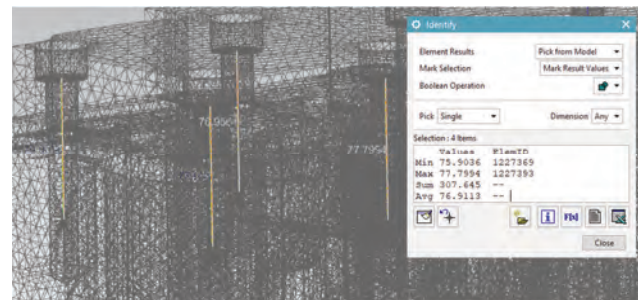


Fig. 13. Axial stress in screws of the cylinder head

The stress in the screws in the integrated main bearing support around the mounting area of the first cylinder amount to 79 MPa at the average. Like in the cylinder head, the stress of other screws reaches an average value of 70 MPa. In case of both screws it is enough to use screws of 8.8 class, having a tensile strength of 800 MPa, and yield strength of 640 MPa.

5. Modification proposals

In the course of the calculations it has been confirmed that the cylinder block in question is not significantly stressed. Therefore, considerations were performed to determine the possibility of increased efficiency of available material. These were performed bi-directionally. The first attempt was made to reduce the mass of the cylinder block. In the second attempt, the maximum pressure in the combustion chamber has been increased in course of the calculations.

Modification of the dimensions of the cylinder block is difficult due to a few reasons. The modifications often necessitate a modification of other elements, which in turn could lead to the need of modifications of other components mounted on the cylinder block. In the end, with slight changes to the thickness of external walls, with modified reinforcements on the cylinder block, and reducing thickness of mounting areas of the crankshaft, the mass of the cylinder block was reduced from 34.17 kg to 31.2 kg, what denotes reduction of materials demand to production of the cylinder block with 0.9.7%. It should be underlined that the

modifications were introduced in areas with the lowest load. The authors are aware, however, that fatigue and thermal stresses were not taken into considerations in presented here paper, the calculations were made for a single stroke of engine operation only. Difference in the manufacturing technology of the block have to be taken into account as well.

To assess a possibility of increasing pressure in combustion chamber, the maximum combustion pressure has been increased to 20 MPa in the mathematical model. The load on the main bearing mounting points has been increased to 76,100 N. After analyzing the results it has been noticed that the deformations of the cylinder walls have remained at the same level as before, in the lower-stressed version. Stresses in supporting areas of the crankshaft were slightly increased, now reaching value 30 MPa at the average and 51 MPa at the maximum. Previously these values were equal to 29 and 44 MPa respectively. In spite of changed pressure acting on the base plate of the crankshaft, the maximum stresses remained unchanged and were still at the level of 1182 MPa. Also in this case the problem connected with singularity of the FEM methods – individual nodes with very high stresses, potentially growing up to infinity – has been noticed. Axial stresses in screws of the cylinder head have increased – at the average now they amount to 82 MPa, and to 83 MPa at the maximum. In lower part of the cylinder block the screws are more heavily loaded. Average stress amounts now to 82.9 MPa.

6. Summary

After completion of the calculations it should be stated that the cylinder block in question was designed so as to avoid any excessive values of the stresses. The stiffness was guaranteed owing to suitable thickness of walls and reinforcements connecting the upper plate with lower part of the cylinder block. Knowing that, more attention should

be paid to connection points with the cylinder head. Considerable values of stresses in nodes where the screws are tightened have been found several times in course of the calculations. Even in case when singularities of the FEM method are neglected in this area, the stresses reach high levels. This dictates a necessity to extend calculations performed up-to-now with analyses, which would take into considerations completely modelled cylinder head of the analyzed engine. The same issue concerns the unified crankshaft main bearing support (i.e. so called bed plate).

Engines of passenger cars are exposed both to changing mechanical and thermal loads. Both the material fatigue and thermal load have not been taken into consideration in this case. However, it seems possible to increase mechanical load of the cylinder block without modification of its dimensions.

The authors consciously did not take into account the influence of temperature in the calculations. However, the presented considerations are the first part of the work on stress analysis of the engine block. In the next part, the authors will carry out calculations taking into account the previously mentioned supplements and thermal loads. The results will be used to assess the possibility of further reducing the weight of the elements and in the didactic process.

In spite of accuracy offered by modern engineering calculation tools using FEM methods, each engine is intensively tested in prototyping phase with respect to its mechanical and thermal strength. It is always necessary to take into considerations material changes arisen during machining operations, in relation to idealized computer model. This problem concerns engine elements exposed to friction force and thermal loads mainly. These processes enable final assessment of the product. The achieved results, however, pave way to a future possibility in creating different versions of the engine based on the same cylinder block, for example a more powerful version of the engine.

Nomenclature

FEM finite element method
CI compression ignition engines
DI direct injection

NVH noise, vibrations, harshness
OEM original equipment manufacturer

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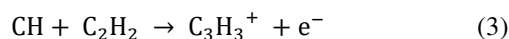
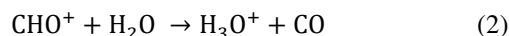
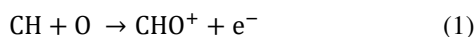
Relations between ion signal and flame propagation in cylinder of a rapid compression machine

Internal combustion engine diagnostics using traditional methods of cylinder pressure signal processing limits the amount of information available about the combustion process. It is necessary to conduct research in order to obtain more precise information – increasing the combustion process diagnosis potential. One such suggestion is the use of an ionization signal and an attempt to link it to the flame development during combustion of gaseous fuels. The article attempts to identify such a relationship using a rapid compression machine due to optical access it provides to the combustion chamber. As a result of the research, the relationships between the ionization voltage (chemical and thermal) of the first combustion phase and the corresponding flame development rates were determined. A relatively high coefficient of determination value was obtained for both relations, which indicates the possibility of obtaining diagnostic information about the combustion process from the ionization signal.

Key words: ion current, flame speed, high speed camera, rapid compression machine

1. Introduction

Chemical reactions occurring during the combustion process in an internal combustion engine are complex and multi-stage. Simplifying the issue, it can be stated that hydrocarbons react with oxygen contained in the air, the product of which is carbon dioxide and water. In fact, other phenomena also occur, including ionization of molecules and atoms. The basic reactions of ion formation in a hydrocarbon flame are [5]:



The ionization process that occurs in an internal combustion engine during the formation of a flame front in a combusted mixture is called chemical ionization [1, 5, 9, 10]. The presence of ions can be detected using an electric field. A flow of current proportional to the density of ions between the electrodes introduced into the volume of the cylinder, to which the potential difference is applied, can be measured. The current thus obtained is called the ionization current. In the spark-ignition engine, the role of the ionization current sensor can be played by the spark plug conventionally used to ignite the mixture [4, 8].

The recombination process is a reverse process to the process of ion formation, and it involves the combining of ions and free electrons. Under equilibrium conditions, the rates of ion formation and their recombination are equal [3].

A useful ionization current signal (Figs. 1 – D, E) as a function of time takes approximately the shape of interpenetrating Gaussian curves [5]. In the phase preceding the occurrence of the chemical ionization signal, there are a number of disturbances (A, B, C) coming from the ignition system. The part related to the flame front formed in the cylinder, the so-called chemical ionization (D) is characterized by a steep rise and a sudden decrease in value. The reason for the rapid changes is the rapid spread of flame in the measuring zone, as well as the presence of mainly free electrons with shorter lifetime [1]. The rate of ion formation

in the rising part exceeds the rate of ion recombination, which results in an increase in signal value [5]. The signal amplitude represents the point at which the rate of recombination is equal to the rate of ion formation [3], while as the rate of recombination increases, the ionization current signal decreases in value.

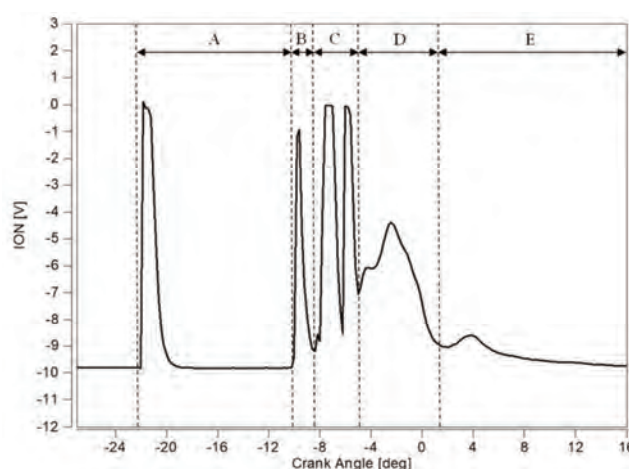


Fig. 1. Real ionization voltage signal with marked phases: A – ignition coil charging phase; B – discharge phase; C – phase of electromagnetic interference; D – chemical ionization phase; E – thermal ionization phase [6]

Assuming that the life time of ions is constant, one can expect a correlation between the nature of the ionization signal waveform and its amplitude and the flame propagation rate in which ions and free electrons are created in the chemical ionization process.

The goal of this work is to attempt to link the ionization current signal to the flame propagation speed in the RCM cylinder using a high-speed camera. Determining the relationship between the indicators will provide information on the possibility of linking the speed of flame development with an ionization signal. Consequently, it can allow to determine the local air excess ratio from the test cycle to the engine work cycle.

2. Method

2.1. Test object

Due to the specific requirements for free optical access to the combustion chamber during the combustion process, the tests were performed using a rapid compression machine (RCM) – Fig. 2. The machine maps the single work cycle of a spark ignition engine to a speed corresponding to the engine speed of 500 rpm. RCM technical parameters are presented in Table 1.

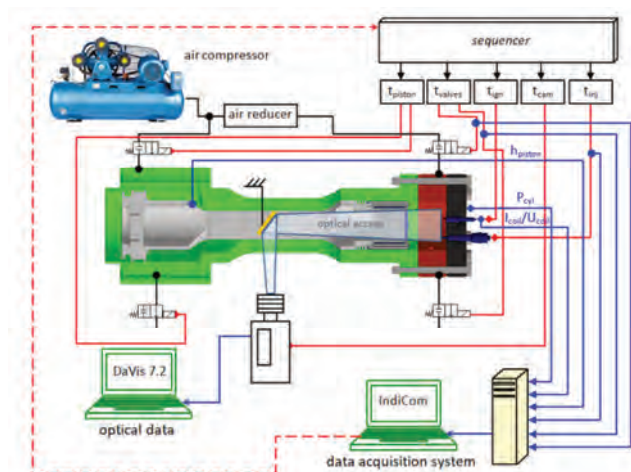


Fig. 2. Diagram of the test stand setup with instrumentation [11]

Table 1. Rapid compression machine parameters

Parameter	Value
Bore x stroke	80 x 90 mm
Compression Ratio	14.7
Speed	up to 500 rpm
Ignition system	spark ignition
Valves system	electromagnetic
Fuel system	direct gas-injection (NG) (electromagnetic injector)
Air system	naturally aspirated

The test object is a mechanical device with electromechanical elements regulating its operation. A series of pneumatic valves controls the piston movement and the supply and exhaust of gases from the combustion chamber. The test stand was equipped with a high-pressure air compressor supplying the pneumatic system (Fig. 3). In order to prepare the mixture in the cylinder, the RCM was equipped with direct gas injection. The spark plug was centrally located in the head. The sequencer with a trigger resolution of 1 ns was responsible for the order in which the subsequent devices were activated.

Data recording was done using the AVL IndiCom data acquisition system. During the tests, the pressure in the combustion chamber was recorded using the AVL GH14D piezoelectric sensor, as well as the ionization current signal using the Mazda Skyactive integrated ignition system. In addition, the following indicators were measured: voltage and discharge intensity at the spark plug, as well as piston paths in a rapid compression machine.

In order to determine the speed of flame development, a LaVision HSS5 high-speed camera was used. The camera uses a monochrome CMOS sensor and allows taking pictures with a frequency of up to 250,000 Hz. The maximum available resolution was 1024 x 1024 pixels. The device is controlled using the DaVis 7.2 computer software.

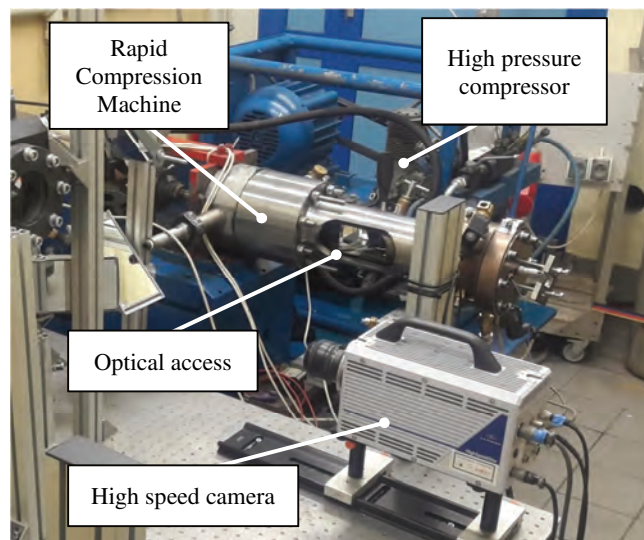


Fig. 3. Test stand for optical research on RCM

2.2. Measurement plan

Studies using a rapid compression machine were carried out in a sequence of 10 measuring cycles under repeatable thermodynamic conditions. The dose of methane delivered to the cylinder was determined based on the previously made injector characteristics: it was 32 mg in each test. The linear speed corresponding to the engine speed $n = 360$ rpm was thus obtained. The spark plug discharge was carried out on average 10 ms before TDC (Fig. 4).

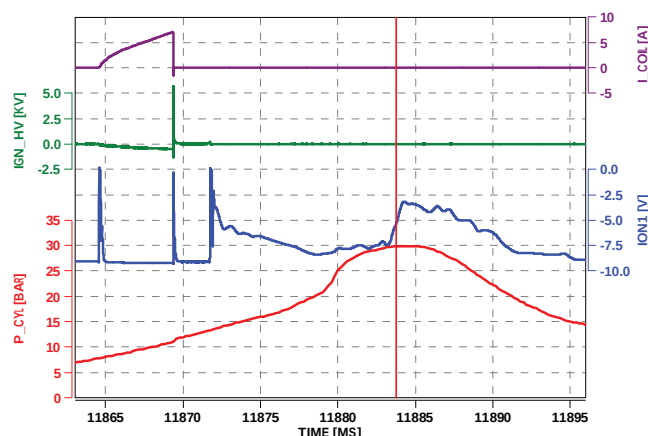


Fig. 4. An example of a pressure signal in a cylinder and electrical waveforms required for ionization signal analysis

The resistance of the spark plug used was 1.6 k Ω , and the distance between the electrodes $d = 0.4$ mm. Other RCM operating conditions are presented in Table 2.

Table 2. RCM operating conditions

n_{av} [rpm]	360
$q_{0\ av}$ [mg]	32
$I_{max\ av}$ [A]	7
$U_{max\ av}$ [kV]	4.6
t_{av} [ms before TDC]	10.6
R [k Ω]	1.6
d [mm]	0.4
camera	$f = 5\text{ kHz}$ $1024 \times 624\text{ px}$

The ionization signal was recorded using the AVL data acquisition system with a frequency of 200 kHz (Fig. 5), while the frequency of taking pictures was 5 kHz. The camera optics were focused on the spark plug electrodes in the cylinder axis (Fig. 2), thus obtaining a flat image exposure. The resolution of the pictures taken was 1024 x 624 pixels.

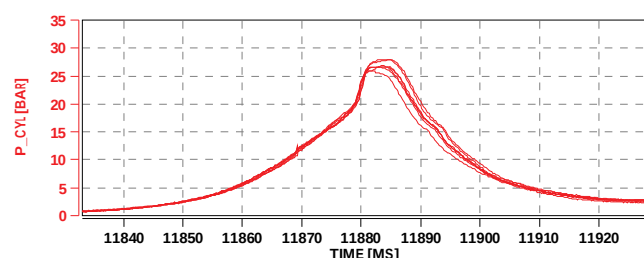


Fig. 5. Cylinder pressure curves indicating the non-repeatability of the work cycles

The analysis of the test results showed a partial non-repeatability of subsequent RCM work cycles [2], therefore the data selection criterion presented in the previous work of the authors [7] was used.

Digital processing of the recorded photos was required to obtain the value of flame development speed. The LaVision camera manufacturer's software – DaVis 7.2 – and an original program written in the internal language were used for this purpose. The image analysis algorithm is presented in detail in [7], and the schematic algorithm in Fig. 6.

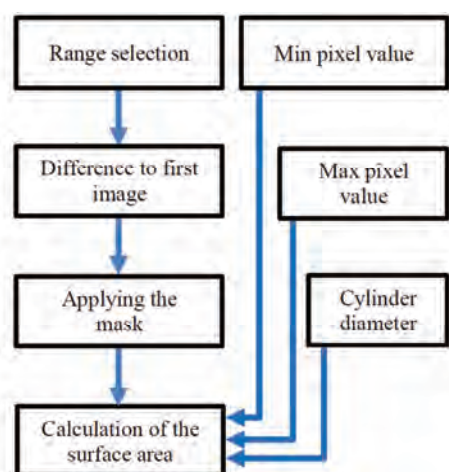


Fig. 6. Algorithm for analyzing images using the DaVis 7.2 program [7]

3. Results

The algorithm presented in chapter 2 was used to determine the surface occupied by the flame (expressed in mm^2) in the combustion chamber of a rapid compression machine (Fig. 7). The flame propagation value expressed in mm^2/ms was calculated for each cycle based on that data. The characteristics prepared in this way were compared with the ionization signal data (Fig. 8).

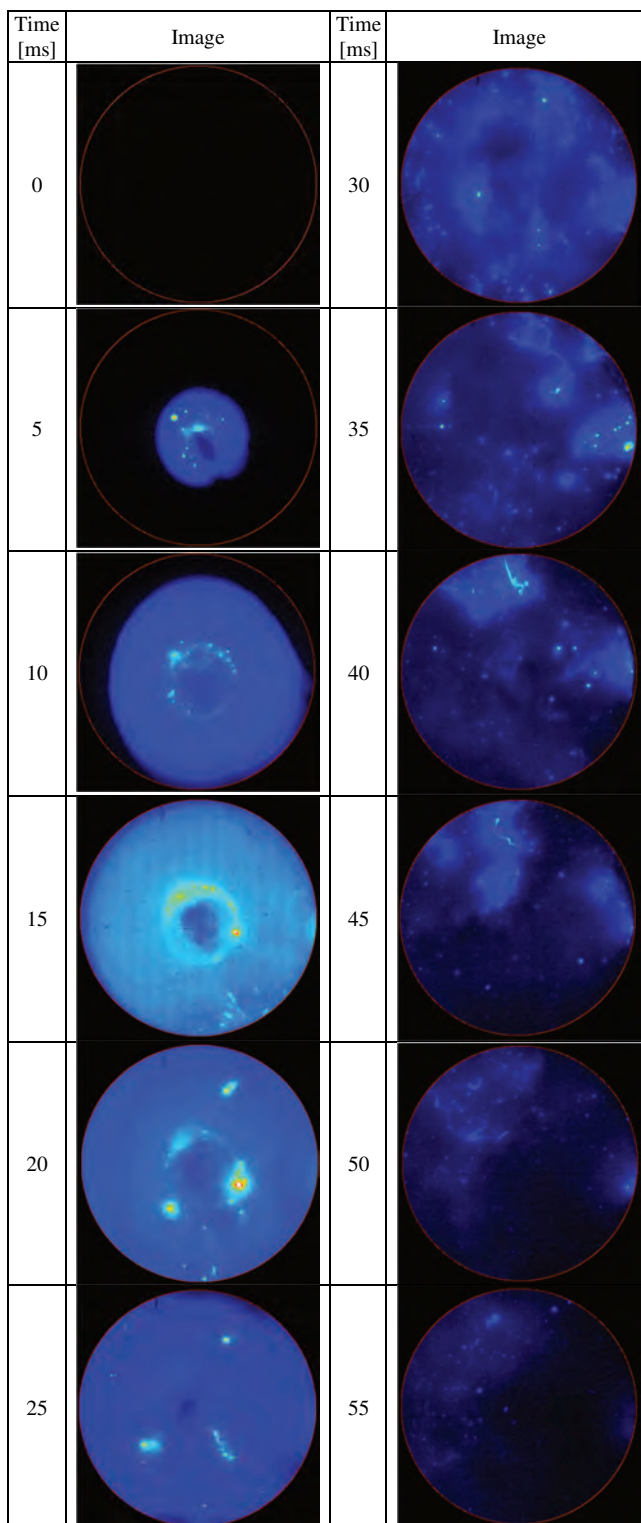


Fig. 7. Sample images of flame development in the combustion chamber

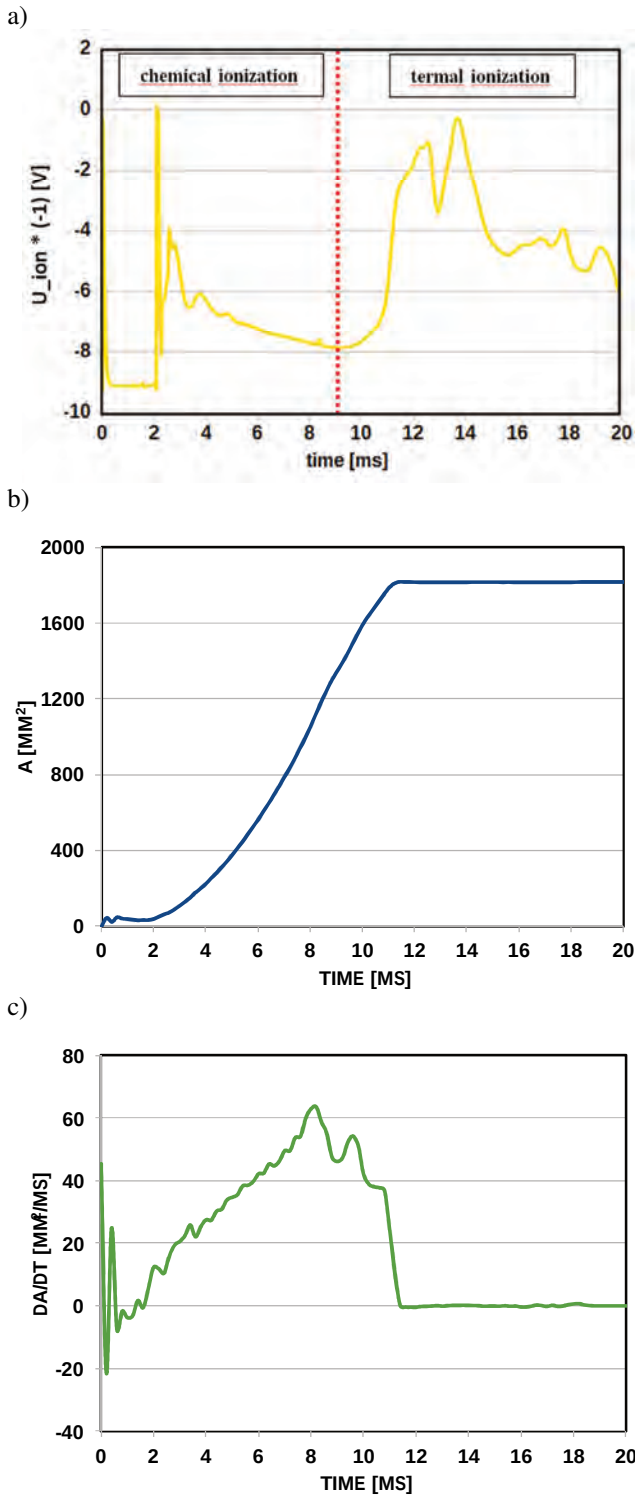


Fig. 8. The ionization voltage function in time (a) and the corresponding waveforms of the flame surface (b) and the speed of flame propagation (c) for a single RCM operating point

One of the criteria for comparing the flame propagation speed and the ionization signal was the coefficient of determination value for the correlation of the maximum chemical ionization signal value and the corresponding flame velocity value. An increase in flame front speed is synonymous with an increase in the rate of ion formation. Assuming a constant recombination time, one can expect an

increase in their instantaneous quantity, and thus an increase in the value of the obtained ionization current signal. To confirm this theory, the maximum ionization voltage (proportional to ionization current) was measured in the chemical phase, which is closely related to the presence of flame between the spark plug electrodes. The flame propagation velocity in the time corresponding to the first ionization voltage measurement was also recorded. Thus, a correlation was obtained with the coefficient of determination reaching a value of $R^2 = 0.65$ (Fig. 9). In addition, it was found that it was possible to achieve better correlation by increasing the number of operating cycles and eliminating reflections in the combustion chamber that interfered with accurate measurement.

Studies, carried out along the main research, on the correlation maximum ionization voltage value in the thermal phase and the maximum flame front propagation speed in the cylinder. It was assumed that the increase in flame speed leads to an increase in the engine thermal efficiency, which means that it is possible to achieve a higher maximum temperature and pressure in the cylinder. These indicators, in turn, affect the formation of ions in the so-called thermal phase. After determining the discussed values, the value of the coefficient of determination for the linear function was reached at the level of $R^2 = 0.68$ (Fig. 10). As in the previous characteristics, it is possible to increase the correlation by eliminating reflections and increasing the number of attempts.

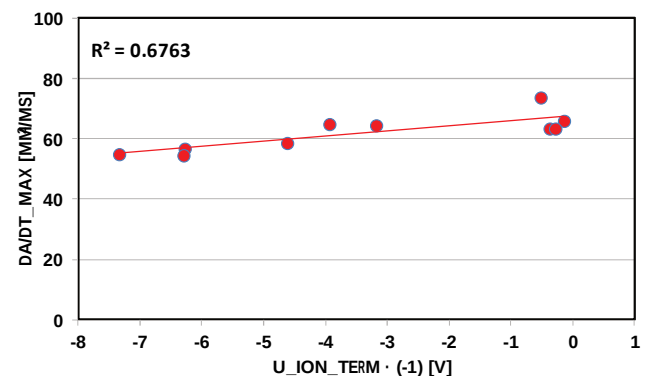


Fig. 9. Correlation between the value of flame propagation speed in the cylinder and the maximum value of chemical ionization voltage

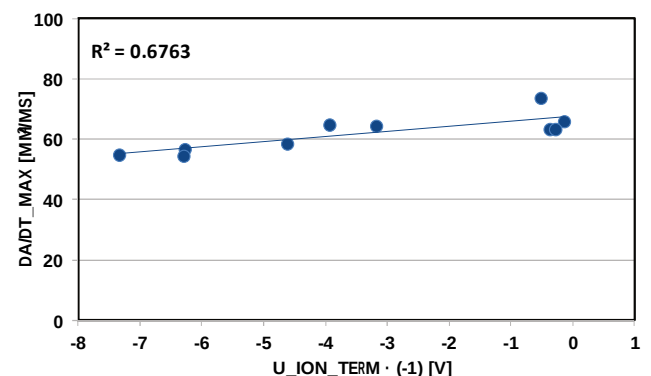


Fig. 10. Correlation of the maximum flame propagation value in the cylinder and the maximum value of thermal ionization voltage

4. Conclusions

Based on the conducted research and analyzes, the following conclusions were presented.

1. A diagnostic assessment of the combustion process using an ionization signal is possible; flame development can be determined indirectly using another diagnostic signal.
2. The relationship between the flame ionization voltage signal in the chemical phase and the flame development rate was obtained with the determination coefficient value of $R^2 = 0.65$, which indicates the possibility of indirect assessment of this quantity; despite many interfering signals, the obtained signal is characterized by high repeatability.

3. The correlation between the thermal ionization voltage signal and the development of the flame of this phase shows a greater correlation determined by the coefficient of determination of $R^2 = 0.68$.

Adjusted ionization voltage signal indicators inside the combustion chamber can be successfully used as diagnostic indicators for an internal combustion engine. They are possible to obtain by much less complicated methods than attempts to determine the flame development in a traditional internal combustion engine. Therefore, systems using the signal of ionization voltage or current are systems that can significantly increase the scope of diagnostics of an internal combustion engine without the need for additional sensors or invasive systems in the combustion chamber of a spark ignition engine when combusting gas mixtures.

Nomenclature

dA/dt	rate of change of flame surface area
U _{ion}	ionization voltage
R	coefficient of determination
A	flame surface area
RCM	rapid compression machine

CMOS complementary metal–oxide–semiconductor

Indexes

chem	chemical
term	thermal

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Biological activity of carbon nanoparticles produced in combustion process

In the recent years industrial applications of carbon allotropes such as carbon nanotubes (CNTs) and graphene have been tested extensively, thus justifying research on the environmental impact these materials have. In the current paper we compare EDS spectroscopy results of a cabin filter used in a car to a filter used in an air purifier inside a residential space. The contaminants adsorbed on the carbon nanoparticles trapped in both types of filters allows for determining of their source of origin demonstrating clearly the dominant role of non-road emissions in Poland. Finally we present the experimental study on the growth of plants on substrates intentionally enriched with CNTs.

Key words: carbon nanotubes; nanoparticles, environment

1. Introduction

The emission of carbon nanoparticles has always accompanied many natural processes such as, for example, forest fires [1–3]. Undoubtedly, the discovery of nanometric particles such as carbon nanotubes and graphene and intensive work on their industrial applications which have intensified especially after 2000 can significantly influence the circulation and prevalence of nanomaterials in nature [4–7]. Regardless of purposeful attempts to use carbon nanomaterials fossil fuel combustion processes are a significant source of their emission. In the studies presented in [8] a strong correlation was found between the concentration of CNT in the lungs of children living in Paris and the presence of gas stoves in their flats. In other works it was shown that vehicle combustion engines are also a source of emission of carbon nanotubes, especially in the case of gas engines [9–12]. These nanotubes have been treated so far as a component of soot, particulate matter containing mainly amorphous carbon and adsorbed impurities. Indeed, if you compare the combustion process in the engine cylinder, you can find similarity to the CVD process, which is the most popular method of growing CNTs. It is also understandable that the nanotubes grown in the combustion process of the piston engine are characterized by a very short length, resulting from the short duration of individual combustion cycles. In studies [13], it was shown that CNT can be intentionally grown on the surface of a cooled metal plate introduced into the engine exhaust gas stream.

The research carried out thus far does not allow to clearly determine the impact of carbon nanomaterials on biological processes, let alone the human body. On the one hand, they imply some extent of carcinogenic effects on animals when inhaled. It can be assumed that the carcinogenic effect is associated with the presence of precursors to the growth of nanotubes or other metal nanoparticles such as iron, nickel, cobalt, which can easily decorate the surface of nanotubes. On the contrary, nanotubes that were cleaned of any metal nanoparticles contamination were successfully used in the purification systems of manned spacecraft [14]. The successes of clinical trials are reported, where CNTs

were deliberately introduced into the human body in place of the missing bone fragments in cases of multiple fractures; CNT serves in this case as a scaffold which initiates the process of bone healing [15]. In a recent study, it was shown that CNT and graphene can be deliberately introduced into the body of spiders, which incorporate them into the web material causing their webbing to exhibit increased strength and durability [16].

The effect of the carbon nanotubes presence in the substrate on the growth of plants and their presence in parts of plants such as leaves or fruits is still not fully understood. Research carried out by Khodakovskaya et al. prove the positive effect of the carbon nanotubes presence on the acceleration of germination and further growth of tomato plants [17]. In other experiments the same authors showed the effect that carbon nanotubes presence has when introduced into the soil when watering tomatoes to their phenotype change. Plants of tomato grown on soil with a CNT addition produced the same amount of leaves, but twice as many flowers and fruits than plants growing in ordinary soil [18].

Other studies carried out on soybeans, maize and beans show a diverse plant response to the presence of carbon nanotubes. The inhibition of small root growth has been proven for all plant species studied. Inhibition of root growth was associated with reduced metabolic activity of root tissue and decreased nitrate uptake, which can be attributed mainly to the smaller root system. The results showed that even under standard growth conditions, largely excluding external factors, plant responses to MWCNT exposure show differences depending on the plant species, but also on the physiological state and stage of development of the individual plants. Detailed analyses made in the case of soybeans have shown that oxidative stress induced by the presence of CNT is the main determinant of inhibition of root growth. Due to the fact that the damage to the plant could be reversed by supplementing the root system with zinc, it was considered that the presence of CNT can immobilize the internal flow of micronutrients. Thus, some environmental phytotoxicity of CNT can be demonstrated along with the negative effect of CNT use in agriculture [19].

In order to verify the positive effect of the carbon nanotubes presence in the substrate, an experiment was carried out cultivating cress on a substrate enriched with carbon nanotubes. Garden cress (*Lepidium sativum* L.) is a species of an annual plant from the Brassica family. In Poland, it occurs under the name of curly cress or pepper grass, which is related to watercress. It is a plant characterized by a high tolerance to various climatic and soil conditions, readily growing with a small amount of irrigation, poor soil and lack of specialized cultivation knowledge. Easy cultivation and tolerance of various climatic conditions gives the opportunity to grow and test it all over the world. The cress can be grown at home and in a garden. It requires the soil or substrate to maintain humidity because the plant is quite shallowly rooted [19]. The fruit of the coleoptera is 5–6 mm long and contains two smooth, ovoid-shaped seeds, pointed strongly in the reddish-brown colour. With dimensions 2.2–2.2 mm long, 1.0–1.5 mm wide and 0.6–1.4 mm thick [20].

Garden cress contains large amounts of iron, calcium, folic acid as well as vitamins A and C. It is characterized by a high protein content (about 25%) including: glutamic acid, leucine and methionones. The seeds contain 22–23% of oil, which used to be extracted for food and technical purposes. Oil of higher purity can be obtained by pressing shelled seeds or seeds with the cover partially removed [21].

Cress leaves have the following composition: 5.8% of proteins, 1% of fats, 87% of carbohydrates, 2.2% of mineral compounds, including calcium (0.36%), and phosphorus (0.11%). There are also trace elements of: iron, nickel, cobalt, iodine, vitamin A, niacin, thiamine, riboflavin, and ascorbic acid. Seeds, in turn, contain mainly alkaloids as well as calcium iron, carotene, riboflavin, uric acid, cellulose, phosphorus, thiamine and niacin. The seed oil contains stearic acid, palmitic acid, linoleic behenic acid, oleic acid, peanut oil, lignoceric acid, benzyl isothiocyanate, benzyl cyanide, sterol and sitosterol [20].

The goal of the research was to determine the effect of the substrate with a carbon nanotube layer on the height of the growing cress seedling.

2. Carbon nanoparticles in the atmosphere

The main sources of carbon nanoparticles emission to the atmosphere are, among others, vehicle combustion engines, industrial processes and fuel combustion in domestic heating furnaces. The last of the sources plays a marginal role in the developed countries of Western Europe, where using solid fuels for heating purposes has been banned for a long time now. Compared to this, Polish villages and cities are unusual, in that household fuel burners using solid fuel are commonly used for heating, low-quality Sulphur containing fuels are burnt often and there is still public and legal consent for general waste incineration in rural areas. Figures 1 and 2 present a macroscopic and microscopic picture, respectively, of a household filter for air purification with pollution accumulated in just a few days of operation during the heating season in a Polish house in a rural area.

Figure 3 is a comparison between the spectroscopy results of the filters shown in the previous figures. Carbon nanoparticles generated in the combustion process adsorb chemical contaminants on their surface, hence the chemical

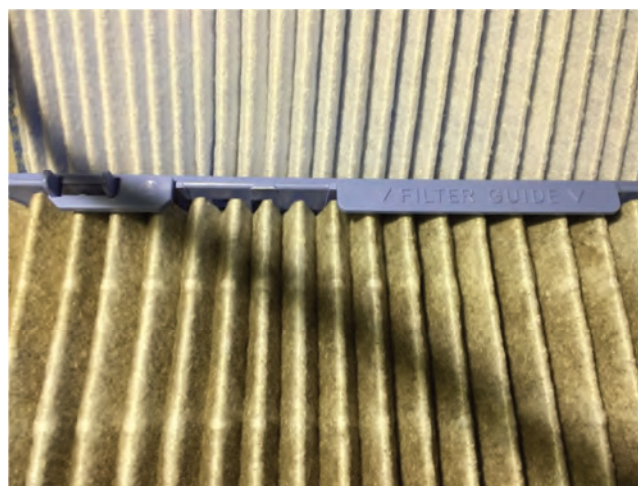


Fig. 1. The filter surface after a few days of operation as domestic air purifier, in the background a new filter for comparison

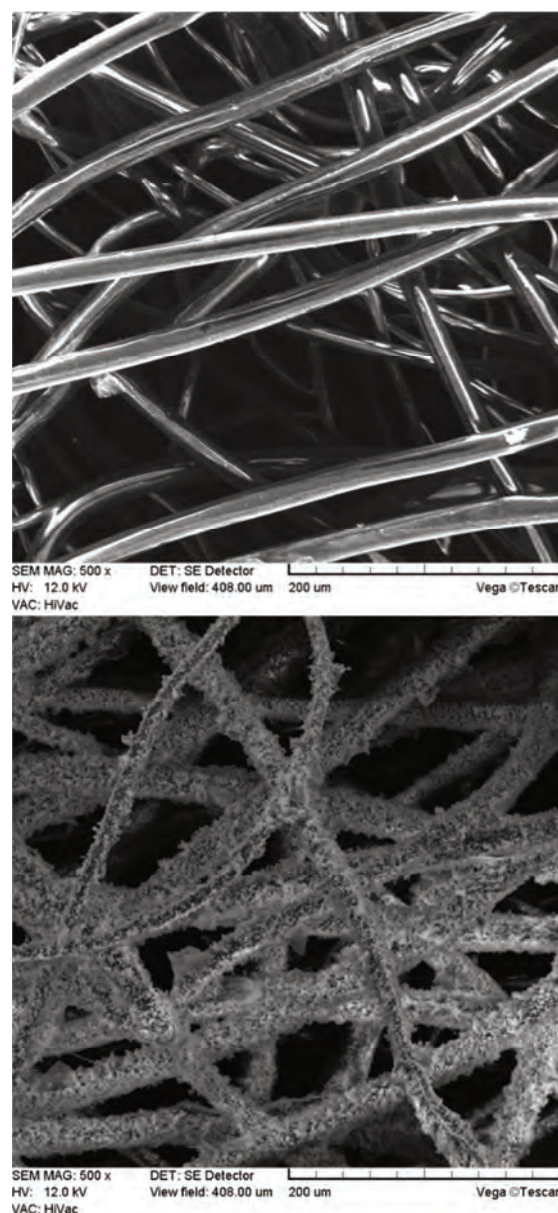


Fig. 2. SEM image of a new air purifier filter (top) and filter after a few months of operation in the winter season of 2018 (bottom)

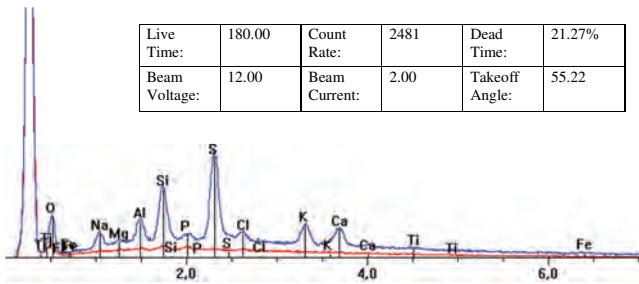


Fig. 3. Comparison of the chemical composition, determined by the EDS method, for the substances on a new filter (red line) and used filter (blue line); filter samples from figure 2

composition of the material collected on the filter allows to determine its origin. In the used filter, apart from the carbon footprint, the peak coming from sulphur has the dominant value. Sulphur has been practically absent in motor fuels for many years, which is why it can be assumed that the accumulated pollution does not come from automotive sources but from the combustion of sulphated fuels in domestic heating furnaces.

3. Carbon nanotubes impact on plants

3.1. Methods and materials

For the experiments, the seedling of cress (*Lepidium sativum*) has been chosen, characterized by low requirements, which allowed to eliminate the influence of other factors on the growth of the tested plants. For the tests, the seeds were selectively checked for any damage or faults. Samples were planted on Petri dishes with a diameter of 10 cm. Two types of substrates were used. The first substrate consisted of a layer of cotton wool and sterile gauze. The second substrate was a layer of cotton wool and non-woven fabric with carbon fibres as shown in Figure 4.

300 seeds were placed on each substrate. The experiment was carried out for a week. Plants were kept at a temperature of about 23°C. Samples were watered daily with 10 ml of distilled water per day for 5 days. The last 2 days of the experiment the plants did not receive any water. After the experiment lasted a week, samples were taken.

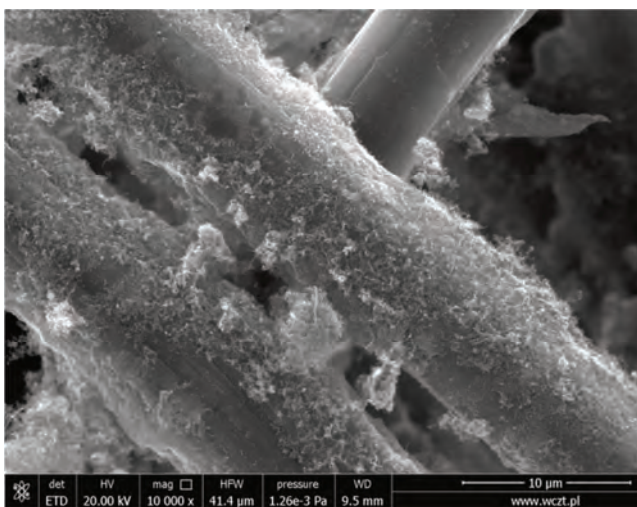


Fig. 4. Hierarchical structure of Nanoveil, a non-woven fabric made of carbon fibers with a layer of carbon nanotubes on the surface

3.2. Results

Figure 5 presents a comparison between the cress seeds samples cultivated on a medium without the addition of CNT and on a medium with the addition of CNT. The visual results indicate a positive effect of the presence of CNT on the intensity of germination and growth of the plant. It seems that the amount of seeds that are germinated is greater in the case of a sample grown on a substrate with CNT. Also, despite the lack of water supply in the last days of the experiment, plants grown on a substrate with the addition of CNT kept the freshness and green colour of the leaves longer. The length of the stalk and cotyledon is much larger



Fig. 5. Cress growth results on an ordinary substrate without CNT (on the top) and on a substrate made of carbon fiber – CNT composite (Nanoveil; on the bottom)

in the case of plants grown on CNT substrate. However, the mechanism of action of CNT on the plant is unknown. Because of the stiffness of CNT-enriched fiber compared to the gauze on which plants were grown without the addition of CNT, it is possible that the water did not penetrate so quickly into the layer of cotton underneath the nonwoven fabric, which resulted in longer moisture retaining the substrate and thus contributed to faster increase and longer freshness of the plant grown on the substrate with the addition of CNT. However, assuming that in the early days when the plants were watered, the humidity of both substrates was similar, it seems that the CNT supplement improved the germination mechanism of plants. Thus, the positive effect of the addition of CNT in the culture medium may be stated by the improved germination of the plant, better water absorption or maintaining a certain stiffness of the stem structure for a long time despite the lack of water supply.

Figure 6 shows the Raman spectra of the cress recorded for three different parts: root, stem and leaf using a wavelength of 514.5 nm. The band's spectrum can be attributed to the following functional groups: bands in the range of $1000\text{--}1150\text{ cm}^{-1}$ are related to polysaccharide vibrations (polysaccharides), $1150\text{--}1300\text{ cm}^{-1}$ are responsible for vibrations occurring in carotenoids, while $1450\text{--}1650\text{ cm}^{-1}$ are associated with carotenoid and chlorophyll vibration. A strong increase of these bands was observed, going from the root part through the stem to the leaf. The CNTs in the Raman spectrum show specific peaks at about 1580 cm^{-1} . Strong Raman signals derived from unsaturated carbon bonds present in chlorophyll and carotenoids do not allow reliable confirmation of the CNT presence in the tissue of the studied plants.

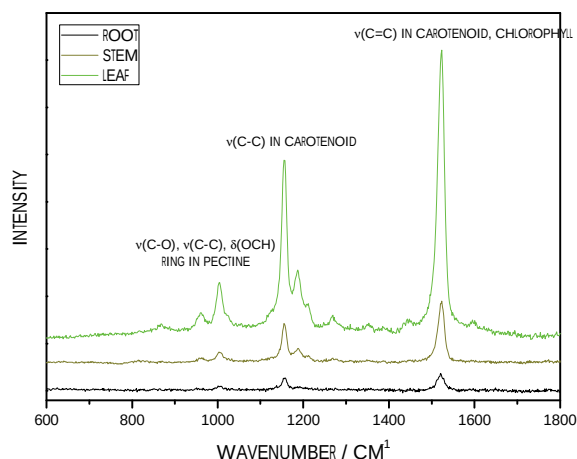


Fig. 6. The Raman spectroscopy results for the plant parts visible in Fig. 5, for the ones growing on a substrate containing CNT

4. Conclusion

The variety of biological interactions of carbon nanomaterials such as TNC and graphene undoubtedly requires additional knowledge and systematization. In this article we presented the results of experiments that show that when implanted into the soil, CNTs stimulate the growth of plants, which at first glance would seem to be a purely desirable effect. It is probable, however, that the presence of CNTs can stimulate the growth of selected organisms, while negatively or neutrally affecting others, which after a certain time could lead to a reduction in biodiversity, with obvious damage to the ecosystem. Our research suggests that CNTs are incorporated into plant tissue, so they can become a component of animal feed resulting in predictable secondary consequences for the environment.

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Analysis of charge exchange in a single-cylinder of spark ignition engine

The article presents the results of tests carried out on a single-cylinder of spark-ignition engine. The charge exchange process in the combustion chamber is very important to determine the possibility of improving the overall efficiency of the engine. Describes of impact modifications engine on open indicator diagrams. Open indicator diagrams show that the higher compression ratio is assumed, the higher the pressure in the system is. First parts of the research describes analysis of charge exchange on not working engine. Results of research presents different open indicators diagrams. Then the pressure results were obtained on the working engine. The possibilities of improving the overall efficiency of the engine by reducing the pumping loss, associated with the replacement of the air fuel mixture, were presented. The article describes the modification of a spark-ignition engine that affects the pressure change in the combustion chamber. The use of innovative methods of regulating the compression pressure gives a lot of positive effects.

Key words: engine, efficiency, compression ratio, pressure

1. Introduction

New design solutions are constantly sought with the purpose of improving the performance of the internal combustion engines. According to the report by the Japanese engineers in the :Research Association of Automotive Internal Combustion Engine: (AICE), research will soon lead to the development of upgrades in the design of internal combustion engines, in which the overall efficiency will be around 50% [6]. However, considerable efficiency of combustion engine does not solve the problem of its poor performance within low and intermediate loads. On the other hand, a direct increase in the efficiency of an internal combustion engine can lead to a decrease not only in terms of the use of hydrocarbon fuel, but also of harmful substances contained in the exhaust gases. According to forecasts by the International Energy Agency (IEA), 70% of all passenger cars and light commercial vehicles produced will be powered by ICEs, and specifically, spark ignition engines. At the same time, it is forecasted that by 2050 approximately 50% of all cars will comprise a traditional ICE.

Internal combustion engines that apply the control of variable compression ratio technology provide an example of a design that combines the traditional solutions and the use of a up-to-date control strategy. Research team from the French company MCE-5 Development S.A. has been concerned with such solutions for several years. Their achievements are significant, as the team reported a 31% reduction in fuel consumption coupled with a 16% increase in engine power output, and a 62% increase in torque [11]. The Toyota team is trying to achieve similar results by synchronizing the combustion engine operating according to the Atkinson cycle in a hybrid power unit as part of the familiar solution already applied in the Toyota Prius car. It also tries to adjust the cycle of the internal combustion engine by applying the variable compression ratio as a result of the design changes, such as in the case when crankshaft axis and cradle axis are offset and the cradle is Pivoted to adjust the compression ratio – VCR Mechanism Assessment. Besides, the innovative trend is based on the solution of Enver company from the USA which focuses on interesting and tangible solutions. Among such designs, it is noteworthy to mention ones that were

developed jointly by Nissan with VW, and currently used in the Infinity passenger cars, not to mention other solutions designed by SAAB and GM. However, despite significant advantages, not all of the above solutions find automotive application, just like the ones that have been achieved in France, as the service life and reliability of the newly designed engines form an obstacle in implementation. Hence, in the study reported by Boretto [7], a MDE (Midsize Diesel Engine) was developed and follows a completely new design approach, including downsizing and downspeeding strategies. It is impossible to list the solutions provided by the VW form with regard to the EA288 3B Generation engines that allows the combustion of lean air-fuel mixtures at low and intermediate engine loads, and this solution achieves the efficiency of the internal combustion engine of above 44%. This is confirmed by the results in which the presented engine reduces CO₂ road emissions in the NEDC test by 10–12 g/km.

Due to the fact that throughout road tests, it is most common to apply the low ranges of rotational speed of the crankshaft and intermediate loads [10], the increase in the energy efficiency of existing ICEs will provide the opportunity to apply them as the principal source of power in vehicles for many years to come. This aspect of the development of modern internal combustion engines is important in the conditions when new solutions focus on the protection of the environment. The reduction of fuel consumption is directly related to the decrease of carbon dioxide emissions to the environment and emission of harmful substances contained in the exhaust gases.

The paper focuses on one of the major trends that provide better efficiency of the internal combustion engine in the range of low rotational speed of the crankshaft and intermediate load. The paper focuses on the potential for reducing losses during the charge-discharge cycle throughout the intake stroke in an internal combustion engine comprising a throttle in the intake manifold.

2. Charge exchange cycle in internal combustion engine

The cylinder filling process takes the interval from the opening of the inlet valves to the instant when they are

closed. In other words, this is the interval in time from the end of the exhaust stroke through the intake stroke and ending at the instant when the compression stroke is initiated. Fresh charge is injected into the cylinder as a result of the instantaneous differential pressure. The course of these pressure changes results from dynamic gaseous phenomena in the inlet and outlet channels as well as the motion cycle of the piston in the cylinder.

The factors involved in the thermodynamic state of cylinder filling include:

- temperature of new charge,
- temperature at the beginning of the filling process,
- pressure of exhaust gases,
- pressure at the beginning of the compression cycle.

The parameters of the exchange of gas and fuel in ICEs constantly change corresponding to the charging cycle in the cylinder. When the outlet the pressure decreases, so does the temperature. To evaluate the process of exchange of gas and fuel in ICEs, it is necessary to determine its thermodynamic parameters such as pressure and temperature and the volume of fuel in the cylinder. When the compression ratio is increased, the area of the heat absorbing surface decreases, which is associated with an increase in temperature [1].

The filling of the engine cylinder is assessed by the volumetric efficiency. The formula for the volumetric efficiency takes the form:

$$\eta_v = \frac{m_r}{m_s} \quad (1)$$

where $m_s = V_{spo} \cdot \rho_o = 1.169 \text{ kg/m}^3$, m_r – mass of delivered charge, m_s – theoretical mass of charge from a specific cylinder capacity and air density.

The increase of the rotational speed of the crankshaft is accompanied by a greater flow resistance, and the charge can even enter the intake manifold. These factors affect the deterioration of the volumetric efficiency. Figure 1 shows the relation between volumetric efficiency and the rotational velocity of the crankshaft.

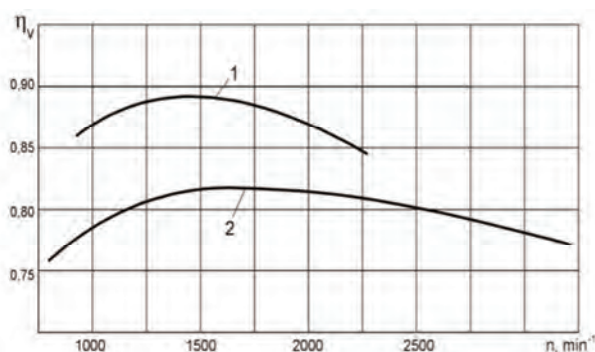


Fig. 1. Dependence of volumetric efficiency on the crankshaft velocity for respective types of engines: 1 – self-ignition engine, 2 – spark-ignition engine [2]

In order to be able to control the charge exchange in an ICE to a greater extent, it is necessary to implement the degree of freedom into the engine control system so that it is possible to control the effective pressure increase in the cylinder. As a result of the continuous increase of the com-

pression ratio in the engine, the performance is not feasible nor cost-effective [2]. In a spark ignition engine, the potential to increase the compression ratio is limited mainly by the occurrence of the phenomenon known as knocking combustion, which can lead to catastrophic failure to the engine. The strength and the occurrence of mechanical losses as a result of too high compression ratios lead to the design of spark-ignition engines with the values of compression ratios in the range from 9 to 14. The notion of engine induction refers mainly to the measurements of cylinder pressure. The adequate use of the induction techniques provides the opportunity of observing phenomena occurring in the combustion chamber at a low cost. The pressure measurement usually involves a full engine cycle.

As a result of changing the compression ratio, is the control the effective pressure in the cylinder is directly affected. The pressure course in the cylinder provides a series of different analyzes. The parameters of the indication can be divided into two categories:

- indirect parameters of the induction,
- direct parameters of the induction.

The difference between direct and indirect parameters means that the direct indication parameters are determined on the basis of the pressure curve derived from a single cycle. Indirect indication parameters require a number of calculations that are needed to determine them.

The control of the effective coefficient of pressure increase in the cylinder is related to the maximum pressure in the cylinder and after averaging it represents the characteristic of the operation of the internal combustion engine in the form of its power, which forms an important characteristic of the engine performance.

Heat transfer takes place between the charge in the cylinder and the cylinder walls and the piston crown. The heat spreads with variable intensity and in a variable direction; therefore, the compression curve taken in the theoretical cycle differs significantly from the compression curve presented in the actual cycle. The curve that describes compression is defined by a constantly changing exponent in a closing range of 1.1 to 1.5 [5]. The following equation defines the parameter of the compression end:

$$p_c = p_a \cdot \varepsilon^{m_1} \quad (\text{MPa}) \quad (2)$$

where: p_a – pressure at the beginning of the compression stroke (MPa), m_1 – mean exponent of the polytropic compression curve, ε – compression ratio.

3. Experimental setup and test object

The test object involved a single-cylinder spark ignition ICE with a displacement equal to 570 cm^3 . This engine included a modified cylinder head design, including:

- addition of an additional bleed channel in the head of the engine to couple the combustion chamber with the environment.
- adding an additional pressure measurement sensor in the combustion chamber.

In this way, the possibility of taking pressure measurements in the combustion chamber was implemented and the possibility of supplying or removing an additional air volume to the charge fed to the engine cylinder during the

intake or compression stroke was introduced. Fig. 2 presents the section of the head with the integrated bleed.

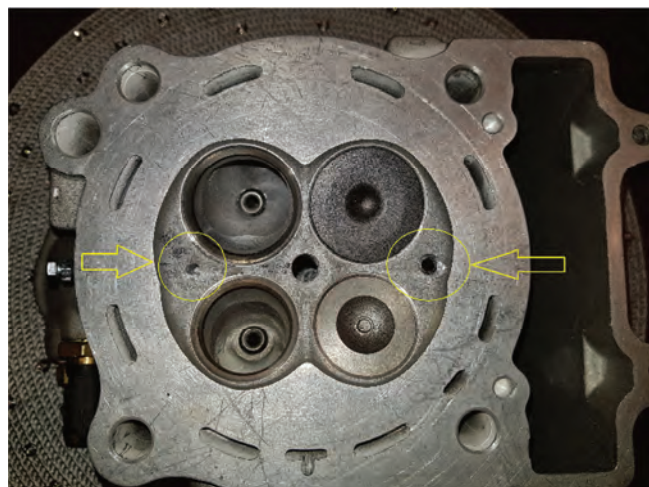


Fig. 2. Combustion engine cylinder head: view of modified head

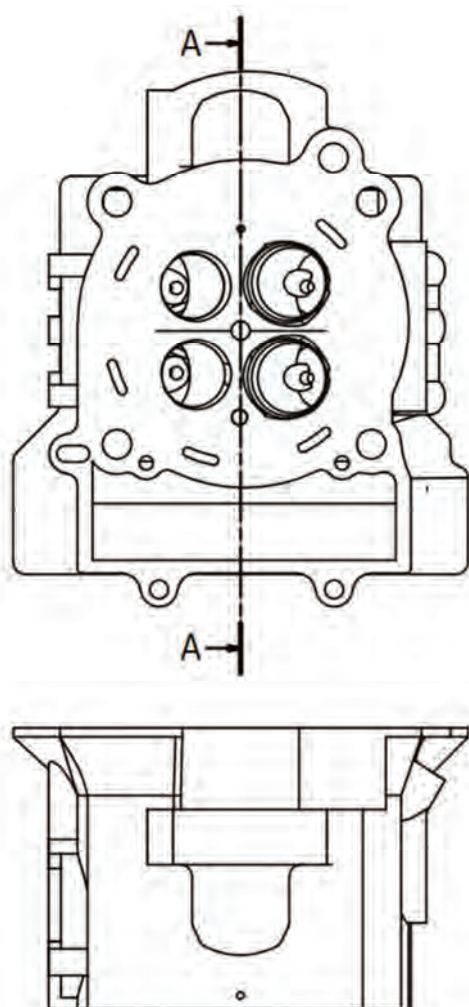


Fig. 3. Model of engine head

Experimental tests have been carried out on a dynamometer comprising an electrodynamic brake that provides conditions for both driving and braking of the investigated engine. A photograph with the engine installed during the

test is shown in Fig. 4, it is coupled with a brake by means of a flexible clutch with non-axial compensation. This solution provided the manner in which axial vibrations of the crankshaft of the internal combustion engine could be compensated.

Throughout the tests, special attention was paid to the process of load exchange in the intake and compression stroke, hence the tests were carried by application of the ICE, but without power input from an electrodynamic brake. The engine is additionally equipped with sensors for the measurement of the following engine operating parameter:

- pressure in the combustion chamber,
- velocity and position of the crankshaft,
- velocity and position of the camshaft,
- throttle position in the intake manifold,
- pressure in the intake system [8].

Due to the fact that this internal combustion engine does not contain a camshaft position sensor, a decision was made to additionally install it (as marked in Fig. 4). The synchronization of the signals from the camshaft position sensor and the crankshaft position sensor provided an adequate determination of individual strokes in the engine. In order to ensure precise and repeatable throttle position in the intake manifold, a control system was designed and assembled to apply Arduino Uno microcontroller.

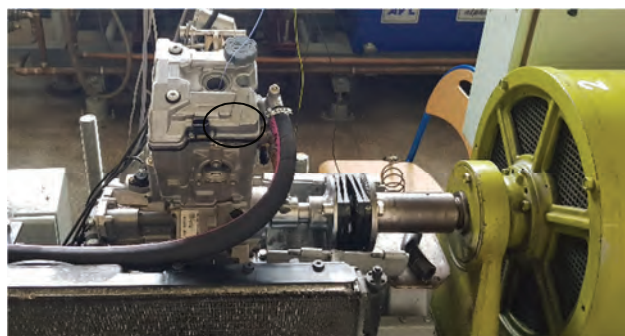


Fig. 4. Experimental setup installed at the dynamometer

4. Experimental study

The increase of the volumetric efficiency in the cylinder was implemented by introducing to the processes of charge exchange of the an additional possibility of feeding or bleeding some of the fuel and air mixture from the cylinder. For this purpose, an additional channel was applied, which included the possibility of adjusting the diameter of the channel leading to cylinder head. Throughout the experimental tests, series of measurements were carried out, which involved changing the diameter of the bore in the cylinder head by attaching previously fabricated orifices. The research comprised several stages, the first of which involved the measurement of pressure in the engine in the condition of the rotational speed equal to 1000 rpm with a closed valve (DCH) in the cylinder head. Two extreme pressure courses inside the cylinder were recorded, for the closed throttle (TPS 0%) and for the open throttle (TPS 100%). Figure 5 presents a engine with closed channel. Figure 6 presents a comparison of the pressure measurements in the combustion chamber.

The next stage of research involved the limitation in the process of air flow of from the cylinder. A non-return valve was installed on the channel to allow air extraction from the cylinder to the environment. As a result of applying this design, during the suction stroke, no air was allowed in the cylinder, whereas during the compression stroke it was possible to decrease the pressure in the combustion chamber. Figure 7 presents the applied 5 mm orifice with a non-return valve and the design excluding this valve, and Fig. 8 contains the results. Involved the introduction of an orifice that provided the possibility of adjusting of the diameter of the 5 mm (DCH) bleed channel leading into the head and pressure measurement in the combustion chamber for the case as before, i.e. with open (TPS 100%) and closed throttle (TPS 0%).

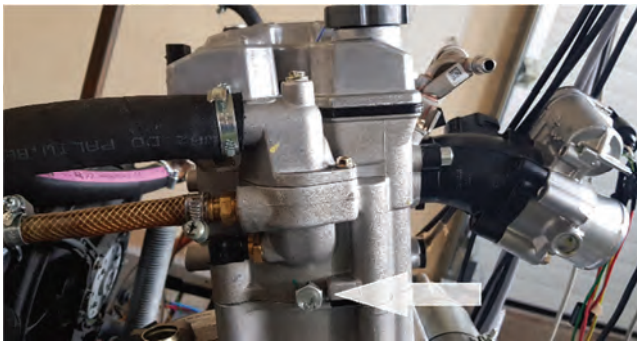


Fig. 5. Engine with closed valve channel

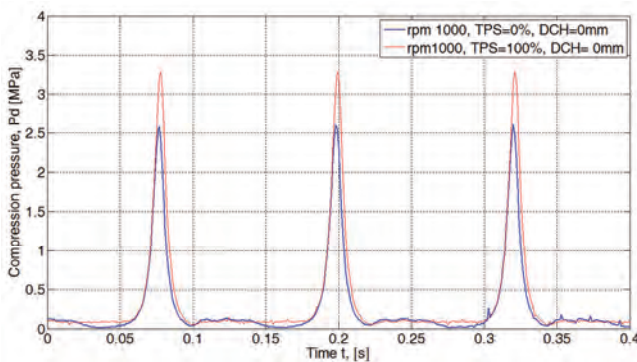


Fig. 6. Pressure curve in the combustion chamber with a closed channel for various throttle opening



Fig. 7. Cylinder head with an extraction channel with a diameter 5 mm and non-return valve

The values and the pressure courses summarized in the table confirmed that it is possible control the value of the effective pressure coefficient in the cylinder. The introduction of an additional suitable extraction valve in the engine will provide the control of the compression processes in the engine. Consequently, it will be possible to develop an engine design excluding a throttle that would significantly reduce pressure losses within the low range of rotational speeds. The use of the extraction channel is described in the patent [P.421255].

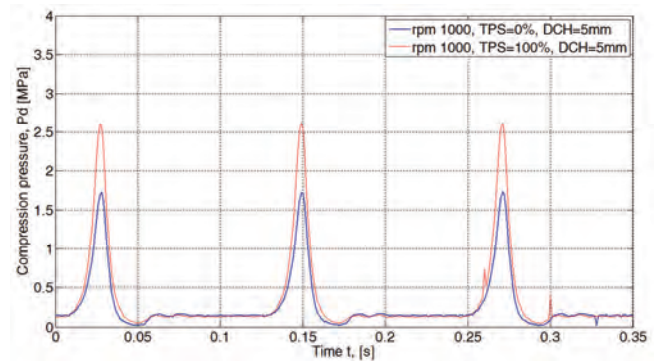


Fig. 8. Pressure curve in the combustion chamber with an orifice with a diameter of 5 mm for various throttle opening

Table 1. Pressure in cylinder

	Max pressure (normal)	Max pressure (5 mm hole diameter)
TPS 0%	2.5 MPa	1.6 MPa
TPS 100%	3.3 MPa	2.5 MPa

5. Extraction valve and further potential of controlling the effective pressure in the cylinder

As a result of the installation of an additional, electronically controlled extraction valve would enable the design and development of an engine without a throttle. In preliminary tests, a solenoid valve was installed in the head of the cylinder of the examined ICE at the outlet of the extraction channel. Due to the need to provide high speed of this valve, an inversion of the design of fuel injector was applied, as it was found to operate at adequate frequency ranges. This solution provided the means to control the rate of the increase in effective pressure for unloaded engine, by adjusting the active engine capacity in the compression stroke. Figure 9 contains a photograph of the head with the installed extraction channel applying a fuel injector.

The control of the extraction valve applied ComactRIO controller (cRIO-9032) with an amplifying system, that was developed as dedicated for the purposes of the present study, and based on the IRF Z44N transistor-transoptor logic was used.

As a result of applying an additional crankshaft position sensor, it was possible to transmit a signal that could be employed for determination of the initiation of the compression stroke. Throttle control was designed and achieved by application of Arduino Uno microcontroller. The extraction valve was controlled by developing the proprietary program in the LabVIEW environment. The course of the signal controlling the bleed valve and the signals of the crankshaft position of the engine and the camshaft is pre-

sented in Fig. 11. TDC position (top dead center) follows 10 impulses of the crankshaft. On the basis of the obtained waveforms, the range of 28 impulses after the TDC of the stroke range was determined as the starting point for the operation of the bleed valve.



Fig. 9. Cylinder head with extraction channel and installed valve

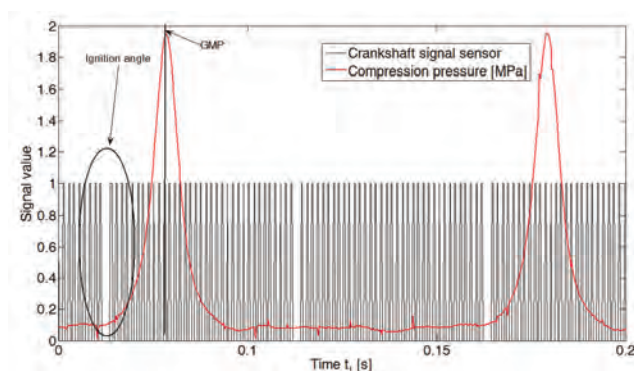


Fig.11 Course of pressure increase in time

In the case of the investigated engine, we have to do with open bleed valve and an open throttle so there is no characteristic pressure drop in the intake stroke (vacuum of pumping loss) and the maximum pressure value assumed in the cylinder is similar to the case of a closed-throttle internal combustion engine. In the further stage of the research, it is projected to develop a bleed valve that will operate at

the frequency of the combustion engine's operation and will be resistant to the conditions in the combustion chamber.

6. Conclusion

The objective of the study presented in this paper involved preliminary research on the possibility of reducing pumping losses and controlling the effective pressure increase in cylinder ICE during the charge exchange in the combustion chamber. Attempts to change the effective pressure increase in cylinder ICE and through it the compression ratio have already been undertaken by companies the automotive sector. The most well-known engine designs characterized by variable compression ratios include: MCE-5 Engine, Scuderi Engine, SAAB SVC, Infinity VC-T. The biggest disadvantage of such designs is associated with their complexity, as each idea implemented in the engine had to comprise a very large number of components that could have been damaged and were characterized by a very high price. The idea that was reported in the paper with regard to control of the effective pressure increase in cylinder ICE does not need a complex design, as it is relatively cheap, durable and the bleed valve can be located in the spark plug or air injector. These ideas have already been filed in the patent office and the proposed solution includes very limited interference in the existing design of the engine.

The research confirmed the assumption that it is possible to perform smooth and effective control of initiation of the compression stroke. The use of a small orifice with a diameter of 2mm resulted in the decrease of pressure in the combustion chamber. It was found that it is possible to apply design changes in order to increase the design compression ratio, e.g. by planning the head, and then controlling the effective pressure increase in cylinder ICE and through it the bleed valve. This solution will allow the engine to adapt to the conditions on the road.

The authors of this paper carry out a number of other tests related to the modification of the engine design, such as resulting from the use of a larger bore size of the bleed valve, application other types of bleed valves to solve the problem with the operating frequency and cooling of the entire system.

Nomenclature

CR compression ratio
BMC bottom dead center
ICE internal combustion engine

TDC top dead center
TPS throttle position sensor
VCR variable compression ratio

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Impact of the use of comfort devices on the exhaust emission from a hybrid vehicle

Vehicles are equipped with more and more devices to improve the comfort of traveling. They are usually powered by electricity generated by the engine, which translates into an increase in its loads and, as a consequence, fuel consumption and emission. However, there is no information about the possible increase in the amount of other harmful components contained in the exhaust gases. Often this result is inadequate to that obtained during the operation of the vehicle, where the obtained fuel consumption is higher. As part of this article, tests were carried out in real hybrid vehicle traffic conditions on the same test route using an analyzer from the PEMS (Portable Emission Measurement System) – SEMTECH DS Sensors Inc. The analysis of gaseous components of exhaust gases together with the exhaust mass flow probe and the GPS system made it possible to calculate the pollutant emission. On this basis, the actual mileage fuel consumption of the tested vehicle was calculated using the road emission of carbon-containing compounds (carbon balance method).

Key words: emission, hybrid, fuel consumption

1. Introduction

The vehicle engine, apart from generating the driving force, must also power the vehicle auxiliary systems. Today, the number of comfort devices, usually electrically operated, continues to increase in vehicles [5]. Many have become standard equipment in most of the produced cars, such as electric windows, and mirrors. Examples of luxury car comfort equipment include, for example: ventilated seats, multi-zone automatic climate control, pneumatic suspension or automatic driving systems. The additional power must be generated by the alternator by charging the engine crankshaft and thus affecting fuel consumption. This, in turn, directly affects the vehicle CO₂ emissions [6, 7]. However, their impact on emissions of other components has not yet been investigated, especially in a hybrid vehicle, where the computer determines the operating point of the internal combustion engine in terms of speed and load.

Increasing the engine load is expected to lead to a deterioration of ecological indicators, and in the case of engines with direct fuel injection, also an increase in the emission of particulate matter [1–3]. Research carried out as part of the article is an investigation on the actual exhaust emissions. In the case of hybrid vehicles in which the engine operates only periodically during vehicle movement, it may be more difficult to heat up the exhaust aftertreatment systems. Temperature is a factor required for their efficient operation, i.e. achieving the light-off temperature and performing oxidation reactions and reducing toxic compounds [8].

In the article, a hybrid vehicle belonging to the F segment, i.e. luxury vehicles equipped with a hybrid drive systems, has been used for research. The research problem is the impact of the increase in energy consumption caused by the vehicle comfort systems on the exhaust emissions. The authors expect increased fuel consumption, but the impact on other compounds is unclear. This problem is caused by the non-continuous nature of operation of the internal combustion engine, which works only periodically in this case which leads to difficulty in keeping the exhaust

aftertreatment systems at their functional temperature and thus enable them to operate efficiently [4]. The tests were carried out by connecting mobile exhaust gas analyzers to the vehicle exhaust system and performing a test drive cycle on the same route in real operating conditions twice, once with comfort systems switched on and once with them being off.

2. Research methodology

2.1. Research objects

The tested vehicle was equipped with a petrol engine with a displacement of 4.0 dm³, power of 404 kW and a direct fuel injection, as well as a 100 kW electric motor [9]. The total maximum power of the drive system is 504 kW and the maximum torque is 850 Nm.

Table 1. Data of the tested vehicles engines [9]

Parameter	Value
Combustion engine type, number and arrangement of cylinders and valves	8-cylinder, 4 valves per cylinder, SI V-engine
Fuel injection system	Direct injection
Displacement	3.998 dm ³
Maximum power	404 kW at 5750–6000 rpm
Maximum torque	700 Nm at 1960–4500 rpm
Compression ratio	10:1
Aftertreatment systems	Three Way Catalyst
Turbocharging	VGT turbocharger with intercooler
Electric motor power	100 kW at 2800 rpm
Electric motor torque	400 Nm at 100–2300 rpm

Due to the hybrid system, the vehicle was characterized by its considerable unladen mass, which amounted to 2310 kg. During the tests, due to the test apparatus, the total weight was approximately 2700 kg. The mileage of the vehicle is 1000 km. Before the tests, the technical condition of the vehicle was checked and the electrochemical batteries were fully charged.



Fig. 1. The tested vehicle with PEMS apparatus

2.2. Used apparatus

The equipment from the PEMS group was used for the measurements. It is a set of analyzers that analyze the exhaust gas sample taken from the exhaust system for the presence of harmful components in the form of: carbon dioxide and monoxide, nitrogen oxides, hydrocarbons and mass and number of emitted particles.



Fig. 2. View of the SEMTECH DS analyzer [10]

Particulate emission measurements from modern vehicles with spark-ignition engines and direct fuel injection are particularly important because of their negative impact on the human respiratory system. Research on the impact particles have on human health lead to the introduction of emission limits. Initially those limits considered only the particle mass, but later also their number, for compression-ignition engines. The limit of the number of particles also applies to vehicles with SI DI engines since September 2017. For this reason, it is required to use two analyzers – one used to determine particle mass, and the other to obtain their number (Fig. 3). The measurement of particulate mass was made possible by the AVL MSS (Micro Soot Sensor) analyzer, based on the PAAS (Photo Acoustic Soot Sensor) method. In this method, solid particles are subjected to radiation of modulated light, resulting in their periodic heating and cooling. This leads to changes in the volume of the carrier gas being tested, acting like a sound wave. In the measurement, vibration-sensitive microphones are used only in a certain range of frequencies and amplitudes. When the air is clean, no signal is detected, while the in-

crease in the number of solid particles in the gas (increase in concentration) increases the sound signal. In order to avoid the formation of a soot condensate, the exhaust gases are diluted (Fig. 3a). The Engine Exhaust Particle Sizer (EEPS™) 3090 spectrometer measures the emitted particle size distribution from 5.6 to 560 nm at the highest available resolution (10 Hz). This allows visualization and testing of dynamic changes of particulate emissions (Fig. 3a).

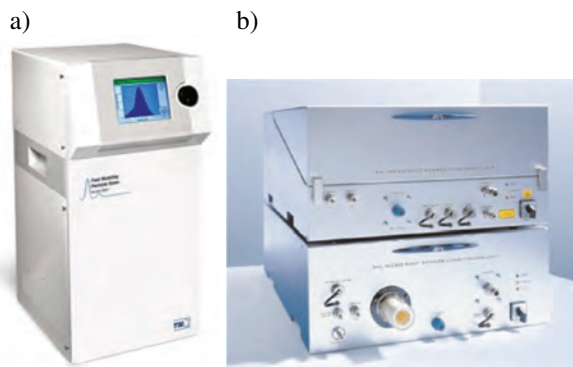


Fig. 3. Equipment used for measuring particulate matter in the scope of: a) number and dimensional distribution - EEPS TSI 3090, b) concentrations - AVL MSS [10]

2.3. Test route

The route used for the study was located in the center of the city of Poznań (Fig. 4). It is characterized by different speed limits and well reflects the conditions of vehicles driving in the urban agglomeration. There are numerous infrastructure elements on it, such as several lanes and intersections with traffic lights. The route was covered twice on the same day, which resulted in similar conditions of vehicle movement and the possible impact of environmental change on the measurement results was minimized.



Fig. 4. View of the route used for testing [Source: Google maps]

3. Results

As part of the research, two trips were made on the same route with the tested vehicle. The first trip was carried out in a mode where comfort systems were not used, i.e. air conditioning, audio devices, seat ventilation and driving assistants. During the drive, the combustion engine worked in a limited range (Fig. 5).

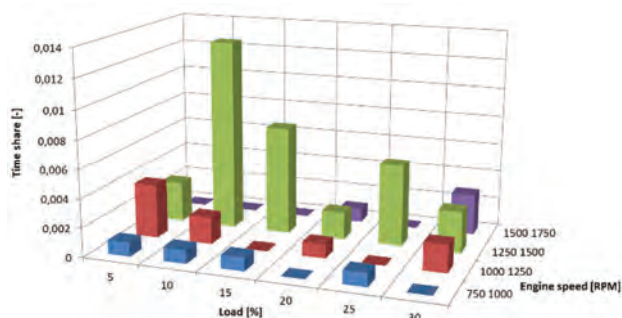


Fig. 5. Engine operating time share according to load and engine speed

The limited use of the internal combustion engine in the range of low rotational speeds and loads results from its considerable power, mostly unused for urban driving and instead the increased use of the electrochemical battery. During testing with PEMS, it is impossible to determine the power output of electric motors. The distribution of emissions of individual exhaust components is shown in Figs 6–10.

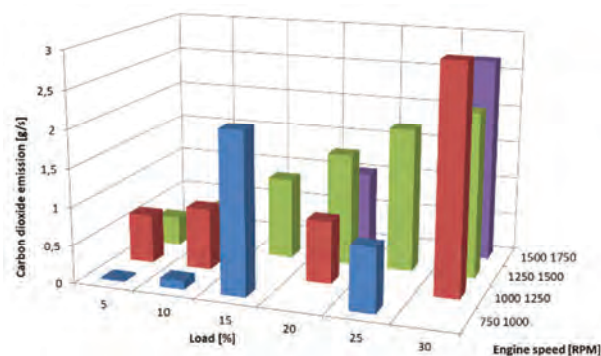


Fig. 6. Carbon dioxide emission according to load and engine speed

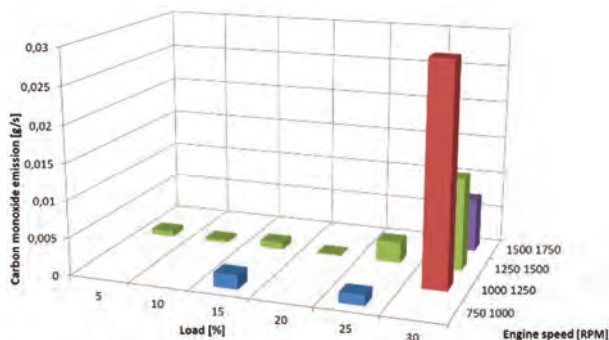


Fig. 7. Carbon monoxide emission according to load and engine speed

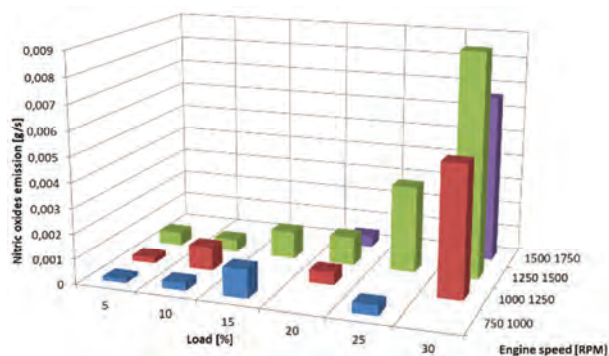


Fig. 8. Nitrogen oxides emission according to load and engine speed

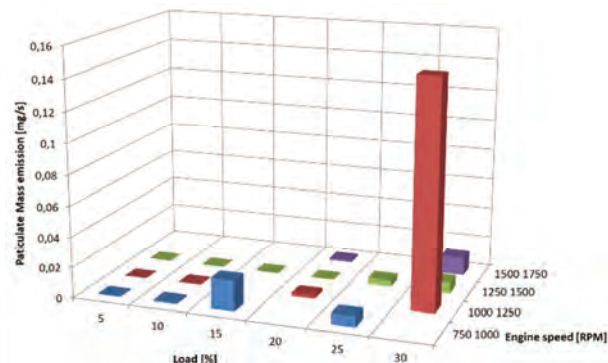


Fig. 9. Particulate mass emission according to load and engine speed

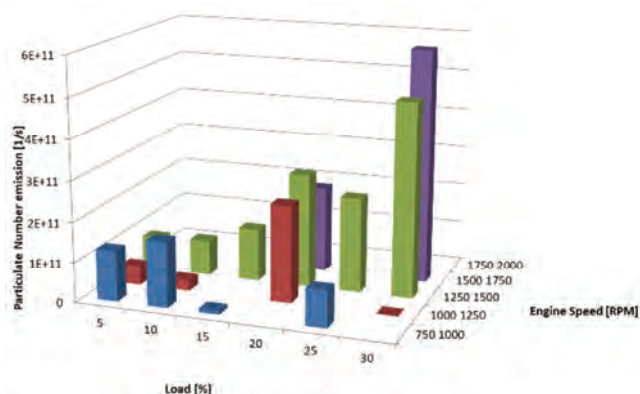


Fig. 10. Particle number emission according to load and engine speed

Carbon dioxide emissions are associated with increasing engine load. The increase is proportional to the increasing load. In the case of carbon monoxide and particulate mass, visible emission peaks can be seen in the range of the maximum engine load reached in the speed range of 1000–1250 rpm. The emission of these components is related to each other and caused by the enrichment of the fuel and air mixture. The results of nitrogen oxide emissions are distributed in a way that is more proportional to the engine load. The second drive took place under the same conditions and with the same battery charge. Already in the case of analysis of engine operating points, differences can be observed (Fig. 11).

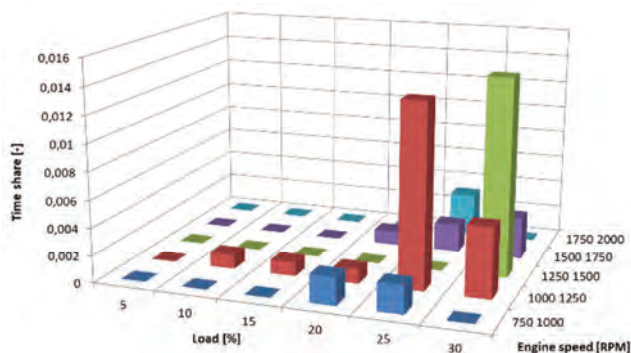


Fig. 11. Engine operating time share according to load and engine speed

The largest shares moved towards higher loads, which is clearly illustrated by the increased demand for energy

caused by the active use of vehicle comfort systems. The emission of harmful components is again presented in the graphical form in Figs 12–16.

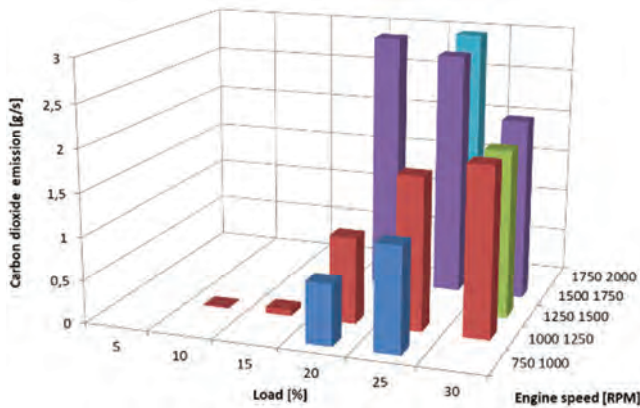


Fig. 12. Carbon dioxide emission according to load and engine speed

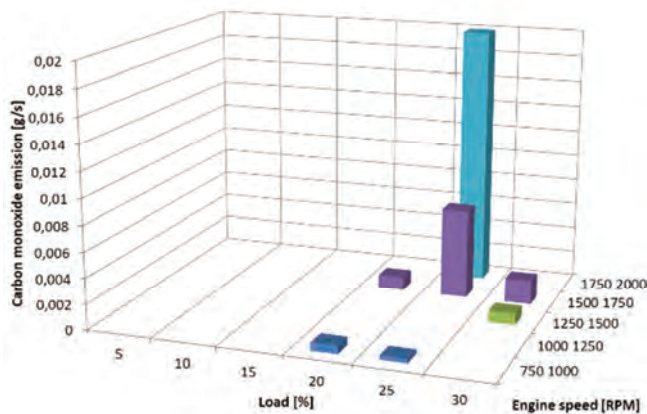


Fig. 13. Carbon monoxide emission according to load and engine speed

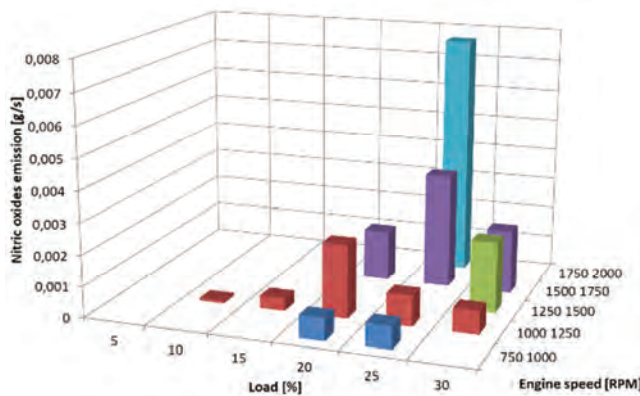


Fig. 14. Nitrogen oxides emission according to load and engine speed

In all the graphs one can notice practically no emission results in the range of up to 15% of the load. Emission of carbon dioxide (Fig. 12) moves according to engine operating points, but during the second test drive the peaks are at a similar level. The carbon monoxide emission is similar (Fig. 13) and obtains similar values as in the case of the first test drive (Fig. 7). The emission of nitrogen oxides for the second drive shows a different character than in the first

drive, but the obtained maximum value of per second emission is closer in value. The biggest differences can be seen in the case of particulate emissions, both in the mass range (Fig. 15) and in their number (Fig. 16). The mass was distributed more evenly, and the number was accumulated in three areas of engine operation. The harmful compounds mass emission results were converted into road emissions (Table 2).

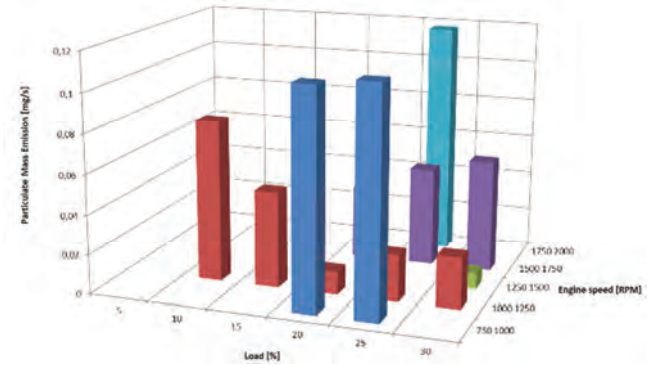


Fig. 15. Particulate mass emission according to load and engine speed

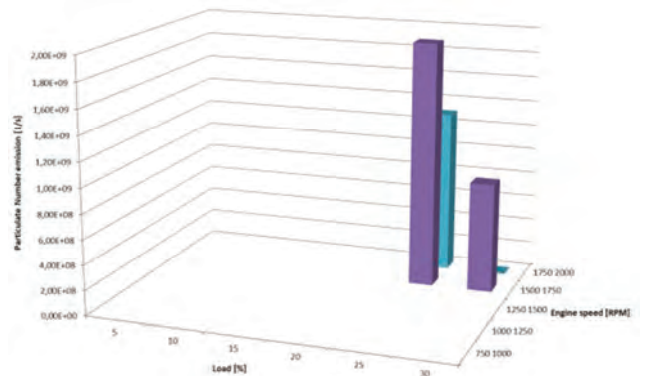


Fig. 16. Particle number emission according to load and engine speed

Table 2. Road emission values for the two drive tests [2]

Emission	Drive 1	Drive 2
CO ₂ [g/km]	11.157	18.671
CO [g/km]	0.039	0.039
NO _x [g/km]	0.02	0.024
PM [mg/km]	0.0114	0.0109
PN [1/km]	1.19E+8	1.75E+9

In the case of emissions of carbon monoxide, nitrogen oxides and particulate mass, the differences are minimal, and their occurrence cannot be demonstrated by activating the vehicle comfort systems. The values of carbon dioxide emissions are proportional to fuel consumption, and thus their increase by over 50% can be observed, however, in the case of the particle number, the value increased by a factor of 12. For this reason, the authors decided to perform an extended analysis of the dimensional distribution of particle size diameters (Figs 17 and 18).

The distributions clearly differ. Emission of large particles dominates in the first drive, and therefore their overall number is smaller. While the second drive (Fig. 18) shows the increased emission of the so-called Nanoparticles that

are the most dangerous for humans. This is caused by increased engine load, which in the case of direct fuel injection systems leads to decrease in the combustion process quality.

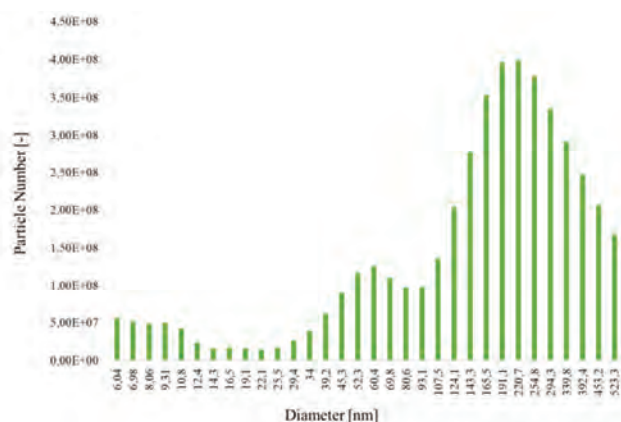


Fig. 17. Particle size distribution in the 1st test drive

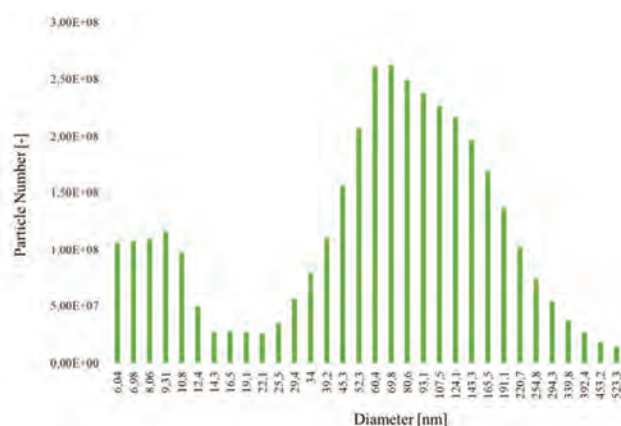


Fig. 18. Particle size distribution in the 2nd test drive

4. Conclusion

New motor vehicles are equipped with more and more technically advanced driver comfort systems, which increase the energy demand related to fuel consumption. As part of the research performed, a new F-segment hybrid vehicle was tested in two test drives: without and with active comfort systems. The obtained results confirmed increased fuel consumption, however, the exhaust emissions remained at a similar level.

However, it should be noted that there were significant differences in the impact of engine operating parameters variability on the characteristics of engine operating time share. The most significant differences in the obtained ecological indicators were recorded for the particle number.

Increased engine load caused a higher emission of the so-called Nanoparticles, which are one of the main problems faced by SI engines today, as has been confirmed in other publications [1–3]. The total particle number emission in comparison to other vehicles was not significant, which was caused by the use of electric drive throughout the majority of the test drive. The latest emission norms introduce tests of vehicles in versions increasingly more supplementary and driver comfort systems, which will better illustrate the actual environmental impact these vehicles have. The direction of further work of the Authors will be the introduction of measurements for electric drive units in order to perform a full energy balance calculation.

Acknowledgements

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Nomenclature

CI	compression ignition
DI	direct injection
PEMS	Portable Emissions Measurement System
PM	Particle mass

PN	Particle number
SI	spark ignition

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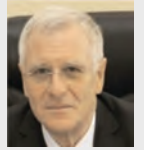
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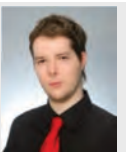
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Methods of evaluating the exhaust emissions from driving vehicles

The type approval tests of vehicles with internal combustion engines increasingly include issues regarding the assessment of ecological indicators in real traffic conditions. This is done with the help of specialized equipment from the PEMS (Portable Emissions Measurement Systems) group. This requires not only a series of test procedures, but also assembly of technically advanced equipment along with the proper preparation of the vehicle's exhaust system. Currently, activities are being carried out to develop solutions for non-invasive assessment of ecological indicators from moving vehicles. The article discusses these types of solutions, at the same time indicating their strengths and weaknesses. Also presented are pollutant emission tests in real operating conditions that will be used to develop a modular exhaust emission gateway. The result of the analysis was to indicate the development directions of methods for exhaust emission assessment from vehicles in motion.

Key words: *emission gate, net emission factor, real driving emissions*

1. Introduction

In 2018, Poland broke air pollution records despite a general downward trend in non-commercial sectors, in recent years emissions in the transport sector have risen again. In the winter months of 2018, air pollution in Polish cities was higher than in the Chinese metropolises famous for smog [11]. An important component of urban pollution is motorization. This is due to the large number of vehicles moving between buildings (often tall buildings) that limit air flow. Therefore, actions are being continuously taken in the European Union and around the world to limit this process. This is served, among others, by the adopted and implemented Sustainable Development Policy [3].

The global trend of motor vehicle tests is focused on the assessment of ecological indicators in real operating conditions [1, 4, 5]. Currently, this is carried out for passenger cars and heavy vehicles in the approval process. This is due to the significant costs of this type of research. On the national roads, the vast majority of vehicles are several or even several dozen years old, which are often characterized by a significant degree of exploitation wear. This results in them generating excessive exhaust emissions, especially of heavy hydrocarbons and particles. In the aspect of annual inspections of technical condition of vehicles, measures are taken to increase the scope of control measurement of pollutant emissions, but there is no definitive methodology that could be introduced for general use. This is mainly due to the difficulty of applying load of the engine at vehicle inspection stations.

For a dozen or so years, devices aiming at non-invasive assessment of pollutant emissions from moving motor vehicles have been developed in various scientific centers around the world. These solutions are often technically limited, and the operating parameters of drive systems are not recorded. As a result, current solutions are insufficient. Such devices are commonly referred to as "emission gates". Their measurement principle is based on optical analyzers or the analysis of the spectra of images taken behind the vehicle. Due to the problem of pollutant emissions from vehicles, it is necessary to further develop such devices.

Therefore, taking into account the significant research and implementation experience of the Institute of Diesel and Transport Engines at the Poznan University of Technology and ODIUT Automex Sp. z o.o. a project is under way: "Emission gate – a modular device for quick assessment of road and rail vehicles emission" (described in chapter 3).

In the studies of ecological indicators of vehicles, the conditions under which they are implemented are very important. This is about road conditions, as well as the nature of roads, weather conditions and ambient (local) air pollution. Determining the exhaust emission characteristics for cities is also very interesting in terms of application possibilities for emission gates. This was the reason for carrying out road tests in the Poznan agglomeration, where the city's emission map was prepared and the ecological indicators of the vehicle with regard to local air pollutants were determined. The research routes adopted included three main communication rings of the Poznan agglomeration, related to the so-called urban communication frames.

2. Solutions for the assessment of emissions from moving vehicles

Solutions for assessing ecological indicators from vehicles in motion have been in the works for decades. However, these constructions are often limited to specific applications (selected ecological indicators, specific type of vehicles or restrictions related to environmental conditions). In some systems, the emission of pollutants is estimated, and for the most part, data on the variability area of the drive system's operation parameters and its thermal state are unknown. For this reason, these devices are characterized by low precision. It should also be noted that they are the subject of patent applications, however in the long term their commercialization is carried out only to a small extent.

Examples of solutions for the assessment of emissions from moving vehicles:

- Patent marked US5498872(A) titled "Apparatus for remote analysis of vehicle emissions" (Fig. 1) [9]. The subject of the patent is a vehicle exhaust gas analysis

device: measuring concentrations and recording levels of NO, CO, CO₂, HC and H₂O. The analyzers used are based on infrared and ultraviolet radiation with a coupled detection unit. The elements are placed on opposite sides of the road. The detector unit divides combined infrared and ultraviolet radiation into separate bands. The ultraviolet ray is directed to a photodiode array in a spectrometer that generates an NO measurement signal. The rotating reflector multiplies the infrared beam over time into many infrared sensors that generate electrical signals that measure: CO, CO₂, HC and H₂O in the vehicle's exhaust. Computer calculates the relative concentrations of CO, CO₂, HC, NO and H₂O on the infrared and ultraviolet beam paths. This device measures only gaseous components without taking into account particulate matter and smoke, NO₂ measurement has also not been taken into account [9].

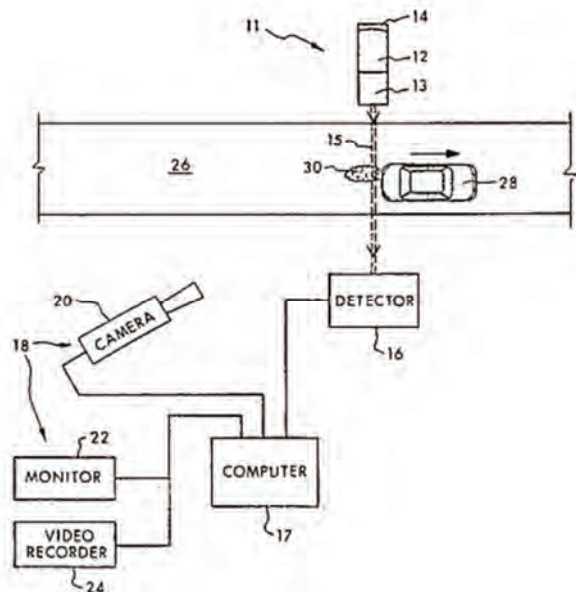


Fig. 1. Operating principle of the device described in Patent US5498872(A) titled „Apparatus for remote analysis of vehicle emissions” [9]

- Patent designated DE4235235(A1) titled “Continuous determination of traffic-generated pollution – employing vehicle-mounted digital signal transmitters which, on passing reference station, identify vehicle to enable pollution assessment” (Fig. 2) [6]. The invention includes a method for determining the intensity of pollutants produced by a vehicle while moving through a reference station. A digital signal transmitter must be used in each vehicle. The signals identify the vehicle type, mass, displacement, fuel type, etc. The reference lines provide the determination of instantaneous speeds. The receivers send signals to the central computer in which the statistical data are classified, where the exhaust emission characteristics related to the above variables are determined. This allows the assessment of ecological indicators of the research object. The video camera provides coarse classification for vehicles not equipped with a transmitter [6].

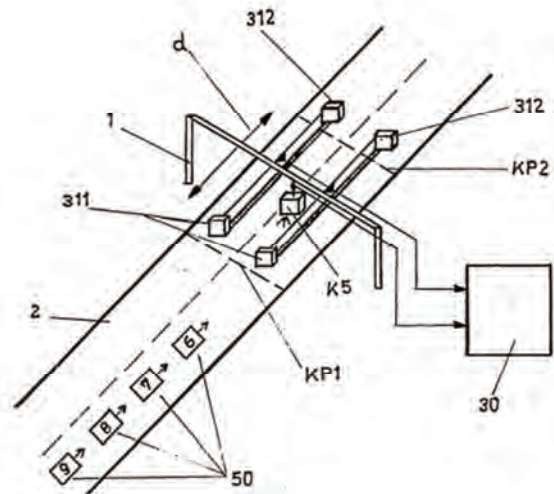


Fig. 2. Operating principle of the device described in Patent DE4235235(A1) titled „Continuous determination of traffic-generated pollution – employing vehicle-mounted digital signal transmitters which, on passing reference station, identify vehicle to enable pollution assessment” [6]

3. Emission gate – a modular device for a quick assessment of road and rail vehicles emission

Emission gates can have a positive impact on the transport sector and the environment. The possibility of testing the actual emission from vehicles with them will significantly affect air purity and reduce the rate of environmental degradation, which is important in the aspect of sustainable development policy. The introduction of gates may affect local emissivity regulations and force them to comply with vehicles of different categories. In an increasing number of European cities, there are laws restricting the entry of older vehicles that do not meet the current emission standards. This applies to the date of manufacture and the fulfillment of certain type of approval standards, and does not refer to actual emissions. Meanwhile, vehicles are often in poor technical condition and thus end up exceeding the pollution limits, especially in the case of PM_{2.5} and PM₁₀ particulate matter. Their level is significantly influenced by solid particles emitted from the exhaust systems. The applicants' research and professional literature confirm that the real emission level differs significantly from the emission determined in the type approval tests and at the vehicle control station (if the latter is performed at all). The impact of motor vehicles on air quality is widely known and more often these observations concern rail vehicles. This is related to a reduction in the life expectancy of people and an increase in disease rates, which mainly concerns upper respiratory diseases and cancers.

The Institute of Combustion Engines of the Poznan University of Technology together with the company ODIUT AUTOMEX Sp. z o.o. from Gdansk, through the National Center for Research and Development, carry out a research project funded by the European Union Development Fund; (POIR.04.01.02-00-0002/18).

The aim of the project is to develop a tool (including manufacturing technology) for measuring emissions of toxic compounds and smokiness of exhaust gases from passing road and rail vehicles. Excessive emission of toxic

exhaust fumes from motor vehicles is a frequent phenomenon and has a significant impact on environmental degradation. The emission gate will enable quick assessment of emissions from vehicles travelling in a given area. It is an important action to protect the environment for current and future generations, which is the basic principle of sustainable development. As part of the project, a modular device based on optical analyzers (NDIR, UV) is developed that can be adapted to the needs of customers. The gate will enable the measurement of: CO, HC, NO_x, PM and smoke opacity in relation to vehicle operation parameters. The properties of the exhaust gases behind the vehicle depend on many factors, which is why the necessity to perform specialized multi-range research works is foreseen. The project includes road and rail vehicles, which will increase the possibilities of commercialization of the project's effects.

4. Measurements of pollutant emissions from a passenger car, taking into account ambient air quality

4.1. Method

The tests were carried out using a passenger car equipped with a plug-in hybrid drive system (Table 1, Fig. 3). It uses an SI combustion unit with a displacement of 1.4 dm³, characterized by a nominal power of 110 kW at 5000 rpm (maximum torque is 250 Nm). The drive uses an

Table 1. Technical data on the propulsion system of the test vehicle

Parameter	Value
Date of manufacture	2018
Displacement	1.4 dm ³
Engine type	SI, plug-in hybrid
Combustion engine power	110 kW at 5000–6000 rpm
Electric engine power	75 kW at 2500 rpm
Maximum combustion engine torque	250 Nm at 3500 rpm
Maximum electric engine torque	330 Nm at 1600–3500 rpm
Engine boost	A turbocharger
Cylinder number	4
Exhaust emission norm	Euro 6



Fig. 3. The test vehicle equipped with the PEMS device during tests in real operating conditions

electric motor, whose maximum power is 75 kW at 1600 rpm (maximum torque of 330 Nm). The electric motor in this unit is powered by a 8.7 kWh battery installed in the floor in front of the back seat. The drive system uses a 6-speed DSG gearbox (direct shift gearbox). The drive system used complies with the Euro 6 emissions standard.

The test drive cycle was carried out in real operating conditions divided into three routes. The first route included a 8.9 km route in the city center of Poznań. The second route covered the city route with a length of 19.3 km, while the third route had the distance of 46.8 km on non-urban roads (Fig. 4). All test sections were carried out on a weekday during the rush hours (afternoon traffic peak). The presented routes ran through the ring communication frame of the Poznań agglomeration.

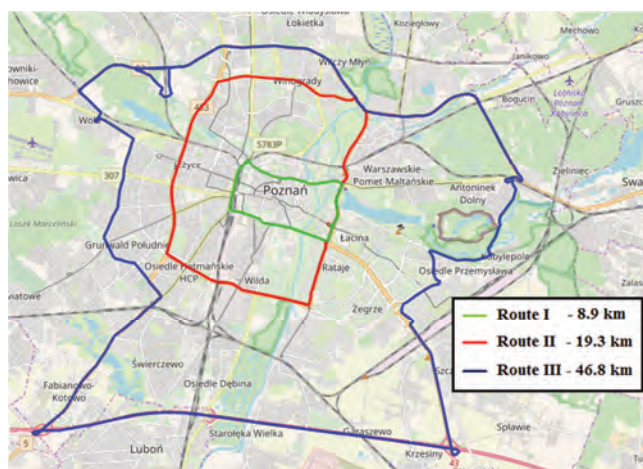


Fig. 4. Test routes in the Poznań agglomeration

Mobile analyzers from the PEMS group were used for measurements. A SEMTECH DS instrument manufactured by Sensors Inc. was used to measure the concentration of gaseous components, exhaust gas flow (CO₂, CO, HC, NO_x) and the oxygen content in the exhaust gas. The measurement was started with calibration. The specific measurement was carried out by taking off the exhaust gases to the analyzer using a measuring probe (gas temperature – 191°C). The sample was taken in through a heated line to prevent the condensation of hydrocarbons. Then the solid particles were filtered from the exhaust gases since they could disrupt the proper functioning of the device. The FID (flame ionization detector) analyzer was used to measure HC concentration. The exhaust gases were then cooled to 4°C. Using the NDUV (non-dispersive ultraviolet) analyzer, NO_x was measured and the content of nitrogen oxides (NO + NO₂) was determined. The concentration of CO and CO₂ was determined using NDIR (Non Dispersive Infra-red). Oxygen concentration measurement was measured using an electrochemical analyzer [7].

AVL Micro Soot Sensor was used to determine the concentration of particles. The exhaust sample was collected by a second independent conduit. The apparatus was equipped with a flue gas conditioning system, and the principle of its operation is based on the photoacoustic method. In the resonant measuring chamber in which the volume of the exhaust gas is located, it alternates heating and cooling.

This causes vibrations of the elastic medium in the chamber. Vibrations are recorded by sensitive microphones as a sound wave. Recorded signals are filtered and amplified, and then converted to PM concentration values [1].

The Axion R/S+ analyzer manufactured by Global MRV was used to assess the emission of pollutants in ambient air. The device measured the concentration of gaseous toxic compounds using an infrared non-dispersive analyzer – NDIR (CO₂, CO, HC) and an electrochemical (NO) and PM analyzer using the Laser Scatter method, in which the particle velocity is measured. The sample of ambient gases was collected near the vehicle's inlet system [2].

Direct data from CAN (controller area network)/OBD (on-board diagnostic) was sent to all the measuring devices, which enabled continuous recording of engine operation parameters. In addition, a GPS location signal (Global Positioning System) was used to determine the speed and position of the vehicle. The control of the apparatus was carried out using a wired LAN (Local Area Network).

4.2. Results and analysis

Due to the hybrid drive present in the test vehicle, different shares of the engine operating time were obtained in subsequent tests (Table 2). During the measurements, the drive system was not interfered with. For strictly urban conditions (route I), the vehicle has obtained over 99% of operating time being exclusively on electric power. It resulted from a large share of downtime. In urban conditions (route II) for the internal combustion engine, the share of operating time accounted for 60%, while in the third section it amounted to 44%. There was a significant load on the vehicle drive system that could not be managed solely by the electrical system and had to be supported by the internal combustion engine. In subsequent tests, maximum speeds of: 60 km/h, 75 km/h and 125 km/h were obtained.

For all three cases of considered measurement routes, a similar share of accelerations was obtained ($a > 0$), which ranged from 30.6%–32.13%. The largest share of vehicle stop equal to 41.52% was obtained for the first section, the smallest for the second 36.53%. While driving on route III, there was a stop at the rail crossing, which significantly increased the share of working time in this area. For the negative acceleration shares, similar values were also obtained in the range of 27.88%–30.54%.

Table 2. The test vehicle operating time share on the test routes in the Poznan agglomeration

Route	Operation time share of the combustion engine	Operation time share of the electric engine	a < 0	a = 0	a > 0
	%				
I	0.33	99.67	30.6	41.52	27.88
II	60.2	39.8	32.93	36.53	30.54
III	44	56	32.13	39.06	28.81

When carrying out the measurements on the test routes, both the ecological indicators of the vehicle and local air pollution were measured. The air sampling probe was located next to the intake manifold inlet. The comparison of the values obtained, including flows in the intake manifold, allows to determine the net emission of pollutants from the

vehicle (Net Emission Factor – NEF). Measurement of pollutants in the air is also extremely valuable in the environmental assessment of a given agglomeration. Performing a series of measurements, in various weather conditions and at different times, allows creating a kind of environmental map. The combination of such data may be useful in the selection of measuring points for the emission gate. In addition, it can be used in the long-term planning and control of traffic (especially freight traffic). An example of a map with the concentration of solid particles is shown in Fig. 5.

The average concentration of PM in the air over the entire test cycle was 0.035 mg/m³. On the other hand, the maximum values reached up to 0.33 mg/m³ locally. Areas with increased air pollution are: the entire route I, the south-eastern part of the route II and the northern part of the route III. The increased concentration of toxic compounds was mainly caused by other vehicles driving in front of the test vehicle.



Fig. 5. Local ambient PM concentration during road tests [10]

The results of vehicle road emissions including local pollutants are shown in Fig. 6. The local concentration was referred to the air flow in the vehicle's intake manifold, thanks to which the net emission of toxic compounds was determined. To determine the emission of pollutants of particular emission factors, the following relationships were used:

$$E_{NEF,j} = E_{RDE,j} - E_{ENV,j} \quad (1)$$

where: E_j – emission of the given compound j ; NEF – Net Emission Factor; RDE – real driving emission value; ENV – ambient pollution.

For the first route, negligible emission values were obtained, which resulted from the prevailing share of driving on electric drive. However, it should be noted that PM emission has negative values of $-0.31 \cdot 10^{-2}$ mg/km. It follows that the vehicle has acted as an air cleaning filter, i.e. when the internal combustion engine was in operation, the ambient air sucked into the engine was characterized by a higher concentration of PM than the exhaust gases released after the aftertreatment systems in the exhaust manifold. The highest values of pollutant emissions were obtained for route II. This was due to the fact that the largest share of combustion engine operation time occurred. On route III, the emission rates were halved on average.

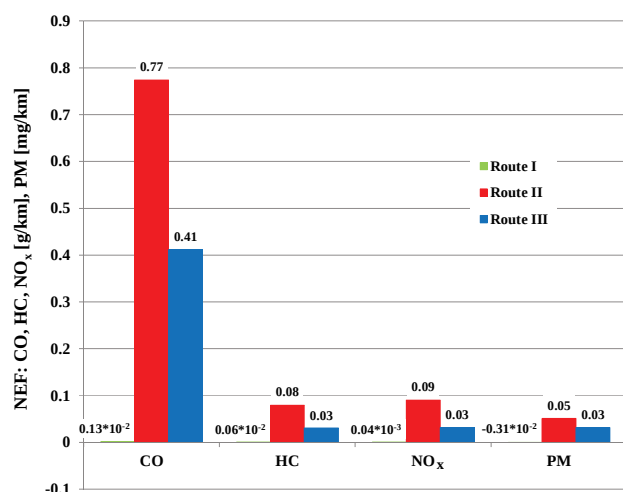


Fig. 6. Net road emission of CO, HC, NO_x and PM from road tests on individual routes

Due to the adopted research methodology, the authors proposed the introduction of a new emission indicator referring to the emission standards of the tested object:

$$CF_{NEF,j} = (E_{RDE,j} - E_{ENV,j}) / E_{limit,j} \quad (2)$$

where: CF – Conformity Factor; limit – pollution limit value j , in which the vehicle was tested for type approval.

CF net indicators have been determined for the presented test vehicle. On the first route, no results exceeded 0.01. As it has already been proved, this resulted from a significant share of driving on an electric motor. For the second route, the highest CF_{NEF} were obtained, with the limit exceeding (i.e. $CF_{NEF} > 1$) being obtained for NO_x emissions. In the third test route, the type approval limits were not exceeded. For gas compounds in the II and III route, the indicator considered assumed values in the range of 0.31–0.8, and for the PM it was in the range of 0.007–0.011.

Taking into account the obtained pollutant emission results, fuel consumption was determined (Table 3). The analysis was based on the carbon balance method for fuel density and toxic carbon-containing compounds. Analogously to the pollutant emission results, the lowest value of fuel consumption was obtained for route I, where the vehicle primarily moved on electric power. For route II, fuel consumption was 6.65 dm³/100 km, while in the third route it amounted to 5.32 dm³/100 km.

Table 3. CO, HC, NO_x and PM net road emissions during road measurements throughout the whole test

Route	Fuel consumption dm ³ /100 km
I	0.008
II	6.65
III	5.32

Taking into account the weights (durations) of individual trips on routes, emission factors for the entire test were determined (Fig. 7). The share for each route was 19%, 26% and 55% respectively. In the whole test, the net road emission of individual toxic components was for: CO – 0.43 g/km, HC and NO_x – 0.04 g/km and PM – 0.03 g/km. Fuel consumption in the whole test overall reached the level

of 4.66 dm³/100 km. The registered emission of pollutants from the vehicle taking into account the locally occurring air pollutants would be higher for: CO by 1.3%, HC by 1.5%, NO_x by 1.6%, and for PM by as much as 46%. It should be noted that it is not possible to add net air pollution locally to the net emissions, because the engine after-treatment systems are characterized by different conversion rates. Therefore, as in the case of PM, it is possible to have the effect of cleaning the ambient air quality.

Referring to the legislative guidelines for the Euro 6 norm, the test vehicle met all the requirements. However, the research did not meet the assumptions of some of the RDE test requirements, and therefore these values cannot be unambiguously compared.

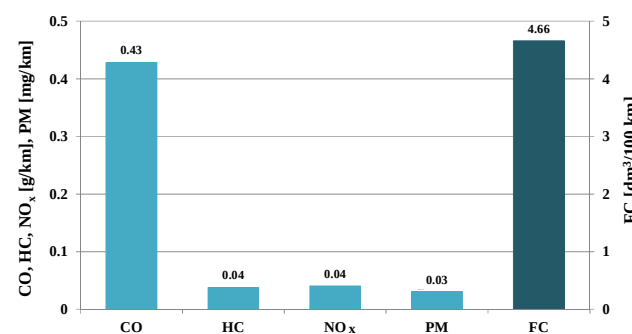


Fig. 7. Net road emission of CO, HC, NO_x and PM in road tests as well as fuel consumption from the whole test drive

5. Conclusions

Exhaust emissions from vehicles in the urban agglomeration pose a serious threat to human health. This is both due to the number of moving cars and the high population density. Therefore, it is necessary to introduce ecological solutions. It is accomplished by implementing a policy of sustainable development, tightening emission standards, intelligent traffic control, etc. Unfortunately, there are vehicles that significantly damage the natural environment – aged, overloaded or converted/repaired not in accordance with the art of engineering, which creates the need for on-going control of the moving vehicles.

Exhaust toxicity tests in real operating conditions are time consuming and cost-intensive. Implementation of one vehicle tests requires at least one day of work and funds of several thousand Euro. For this reason, measures have been taken to assess the emission of pollutants from moving vehicles in a non-invasive way. This is to be implemented by the use of emission gates, which will carry out the assessment of ecological indicators and vehicle operation parameters. The works are carried out as part of the project: "Emission gate – a modular device for a quick assessment of road and rail vehicles emission" within the Smart Growth Program (contract No. POIR.04.01.02-00-0002/18).

To build the gate, it is necessary to perform a series of preliminary tests, mainly vehicle tests in real operating conditions. The article presents hybrid vehicle tests in the Poznan agglomeration, including local ambient air pollution. It allowed to create the city's emission map, define the net ecological indicators of the vehicle and indicate the directions of further activity.

In the route located mostly in the city center the tested vehicle moved almost exclusively using the electric engine, which was very beneficial in terms of reducing the local environmental impact. In some cases, the vehicle acted like a filter because the intake manifold air had a higher concentration of pollutants than the exhaust gases. This type of phenomenon was observed mainly for particulate matter. Thus it can be concluded, that there is a need to continue to measure not only their concentrations, but their dimensional distribution and number.

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Nomenclature

a	acceleration	LAN	local area network
CAN	controller area network	NDIR	non-dispersive infrared
CF	Conformity Factor	NDUV	non-dispersive ultraviolet
DSG	direct shift gearbox	OBD	on-board diagnostic
FC	fuel consumption	PEMS	portable emission measurement system
FID	flame ionization detector	RDE	real driving emissions
GPS	global positioning system	SI	spark ignition

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Problems of exhaust emissions testing from machines and mobile devices in real operating conditions

The article presents selected problems concerning tests of toxic exhaust emissions from engines of vehicles, mobile devices, engines used in gardening and construction works. For this group of engines, this type of testing is carried out primarily in laboratory conditions, however, variable load conditions during actual operation justify efforts to determine the level of emissions in real operating conditions. Research in real operating conditions for different engine categories becomes increasingly more standard, in this aspect, it is necessary to verify the state of knowledge and methodology in the aspect of testing engines of machines and mobile devices. The research performed by the authors was the first of this type using PEMS equipment, hence some conclusions and observations can be made from the obtained results. At present, the type approval procedures for the engine group in question do not account for emissions tests in real operating conditions, while also missing any description of a standardized research methodology. In addition, the engine group in question is not really suitable for testing in real conditions. An introduction to research aimed at developing an exhaust emission test methodology in real operating conditions of mobile device engines was included in the article.

Key words: *emission, combustion engines, PEMS*

1. Introduction

The continuous development of the construction, agricultural and gardening industries proportionally influences the growth of hand operated devices that facilitate and significantly accelerate both the simplest and the most complicated activities. In addition, there is more and more automation, which forces the development of industries, dealing in the production of handheld devices and all kinds of machinery and equipment, which also belong to the segment of non-road mobile machinery. In practice, their availability on the market and easy operation means that we have a wide range of devices equipped with small engines with a net power of not more than 19 kW. The operation of engines used for vehicle drives and devices with various purposes usually takes place in states different from the optimal ones. Variable ambient conditions, high frequency of starting cycles and operation in a wide range of rotational speeds, as well as a wide range of engine loads, make it difficult to achieve a low level of harmful emissions. From the point of view of the emission of toxic compounds, the most unfavorable and unexpected is the operation of the engine in transient states, including in the period of cold start-up and during heating. An engine that has not reached the rated operating conditions, particularly for the temperature parameter, usually has a high concentration of carbon monoxide (CO) emissions as well as hydrocarbons (HC) at a slightly reduced level of nitrogen oxides (NO_x) emissions. Additionally, for the group of CI engines, an increased issue of particulate matter (PM) is also a very important problem.

The current state of legal regulations in the field of homologation and toxicity tests does not impose on manufacturers and users of manual devices using small engines the obligation to subject the operated machines to emission control tests. This is a legal gap, which includes a large group of single-cylinder engines with spark ignition and compression ignition, used to drive small devices. Literature

analysis shows that no exhaust emission standards can negatively affect the pollution of the natural environment, because the increase of toxic exhaust emissions in such engines occurs as a result of their excessive use [4, 5]. In this type of devices, it is the high rotational speeds of the crankshaft of the engine and the power generated in this range during the work cycle, which guarantees a large expenditure of work and short cycle times, ensure perfect engine preparation for operation even in difficult conditions causing large changes in amplitude load. Hand operated equipment engines perform their tasks to a significant extent, with the participation of a working time representative of the external characteristics. Such nature of their work is burdened with a high probability of occurrence of emission changes caused by improper engine operation. This happens due to the wear of its elements and changing the settings of certain parameters. One of the other problems that occurs in spark ignition engines is the need to use an oil fuel mixture in the combustion process. Too high oil content in the fuel volume causes excessive emission of toxic compounds.

Hand machines are most often equipped with a SI two stroke engine. The simpler construction and the more power to weight ratio of the engine compared to the four stroke engine results in fitting of two stroke engines in these devices. In terms of design, they differ from engines mounted in vehicles. These differences result from the specific working conditions of the machines in which they are mounted [1]:

- working at 35–45° lateral inclination;
- work at longitudinal tilt 40–45°, in special applications 60°;
- working conditions: atmospheric pressure, temperature, humidity;
- high variability of external loads;
- adaptation to cooperation with a hydrokinetic transmission or a clutch – converter;
- special designs of air filters to work with high dust;

- installation of silencers in the exhaust system with a spark absorber;
- relatively simple service, especially easy access to consumables (filters, drain plugs for operating fluids).

Despite so many ways to reduce toxic exhaust emissions, it is difficult to reduce them because of the need to maintain a relatively low weight of the device. The need to manually operate a device equipped with an internal combustion engine together with its equipment adapted to perform a specific activity limits the installation of exhaust gas cleaning systems. The presented situation motivates to take steps enabling state-technical assessment of combustion engines in devices with such a wide application, whose population is growing at a significant pace, it seems inevitable that procedures enabling control of emissivity for this group of non-road machines and devices will be unavoidable.

2. Methodology of research

The ecological aspects of internal combustion engines due to the emission of pollutants are strongly determined depending on the conditions of the work performed. It is crucial to correctly interpret problems related to the formalization of the suitability of the conditions of use and the development of an engine test method to assess them. The methods by which this type of information can be obtained are constantly sought for. Standard acquisition of time density characteristics for working engines is carried out during measurements on stationary engine test benches [6]. However, these tests allow only the information to be obtained at fixed work points, which significantly limits the acquisition of a lot of information. Therefore, the determination of representative work points for the category of drive units intended for the operation of non-road mobile machines is in many cases impossible to achieve. Proposing a universal procedure that allows the measurement of a reliable emission level is associated only with the use of a test method allowing the registration of parameters under conditions of actual use [2]. Previously, measurements of this type were limited mainly by the research facilities allowing the work to be performed only in laboratory conditions. The emergence of a method based on the possibility of using PEMS (*Portable Emissions Measurement System*) equipment, gives a new possibility of connecting for research purposes analyzers to small mobile devices equipped with internal combustion engines and adapting it to perform measurements in difficult operating conditions (Fig. 1) [3].

One of the devices enabling the implementation of measurements in real-life conditions is the Axion RS mobile analyzer (Fig. 2) from the company GLOBAL MRV belonging to the PEMS type analyzer group. It is a unique combination of measurement and registration of the following parameters:

- concentrations of CO and CO₂, NO_x = NO + NO₂, THC and O₂;
- determining the mass flow of exhaust gases;
- working conditions – atmospheric pressure, temperature, humidity;

- position and speed of the tested object – GPS (*Global Positioning System*);
- engine crankshaft speed.



Fig. 1. Measuring system in real operating conditions



Fig. 2. Mobile exhaust gas analyzer Axion RS [8]

Table 1. Technical parameters from the analyzer device [8]

Gas	Measurement range	Accuracy	Resolution
HC Propane	0–4000 [ppm]	±8 [ppm] abs. or ±3 [%]	1 [ppm]
CO	0.00–10.00 [%]	±0.02 [%] abs. or ±3.00 [%]	0.01 vol. [%]
CO ₂	0.00–16.00 [%]	±0.30 [%] abs. or ±3.00 [%]	0.01 vol. [%]
NO	0–4000 [ppm]	±25.00 [ppm] abs. or ±4.00 [%]	1 [ppm]
O ₂	0.00–25.00 [%]	±0.10 [%] abs. or ±3.00 [%]	0.01 vol. [%]

An important problem of modern equipment is their limited scope of measurement resolution. The level of detection of concentrations of gaseous exhaust components makes it impossible to study in a wide spectrum, which in

the case of engines not equipped with any exhaust gas cleaning systems as well as those operating in the work circuit, characteristic for two-stroke units, enforces the use of the most modern solutions that give the possibility of diluting the exhaust sample. In order to extend the measurement possibilities in conditions of very high concentrations of gaseous components it is necessary to use an additional system operating as an exhaust diluent (Fig. 3), which in a given dimensional range and dilution tunnel allows for preparation of a sample for analysis in a measuring track. Such equipment allows the use not only in stationary conditions, but also in dynamic conditions, e.g. during engine start up, as well as in transient conditions between dynamic load change.



Fig. 3. View of the operation of the sample diluent tunnel [9]

One of the solutions is a device available from the TSI manufacturer Rotating thermal disk (Fig. 4). Particularly suitable for sampling, diluting and conditioning of exhaust particles from diesel and sparking engines. In addition, it is equipped with a thermal conditioner that can be heated to 400°C, to eliminate nano drops that may have formed during the dilution process. It also has the ability to function functions in the source (exhaust pipe, tunnel or stack) and thermally conditions the particles to keep them accurate for measurement.



Fig. 4. TSI gas diluter [9]

3. Measurements under real operating conditions

In field trials, the preparation of a test device is also a very important element. The key issue is to adapt the test facility in order to be able to mount a group of sensors enabling the determination of the mass value of exhaust gas flow rates. This connection requires maintaining tightness during testing so as to eliminate dilution by atmospheric air.

It is required that the pressure sensor is mounted in a vertical position in order to avoid disturbances in the measurement due to the possible occurrence of condensation of the fuel-air mixture in the intake system. The length of the connection between the flow meter and the end of the engine exhaust system must be relatively short to prevent unnecessary pulsation. The next stage is the construction of a connection enabling the registration of the rotational speed of the crankshaft during the operation of the device. Exemplary methods of application of the apparatus for performing tests under real operating conditions are shown below (Fig. 5).

Before carrying out measurements, the issue of powering the measuring devices should be solved. Under test conditions of tested objects the apparatus needs energy to operate, therefore it is necessary to use an available source from the installation of the device under test. If the tested object does not have its own power supply, or does not generate a minimum value of 12 V, then an additional power generator is needed. In the first stage, the analyzer should be conditioned, the leak test of this connection should be carried out and during the measurements it should be monitored if its tightness is maintained. This has a significant impact on the elimination of possible potential measurement errors. Before starting the work of the object, the registration of defined parameters starts. In order to get a full description, which will enable a complete analysis of the share of working time at individual points, a test should be carried out according to a pre-proposed schedule:

- starting up the device and conditioning during the normalized engine crankshaft speed for idling;
- lower crankshaft speed for generating loads below < 50% of maximum power;
- dynamic changes in the full speed range and load values characteristic of the actual operation conditions;
- higher value of crankshaft speed for the generated load below > 50% of maximum power for a given structure;
- minimum engine crankshaft speed at which maximum torque is possible;
- in the case when the device is self-propelled, registration of parameters during the travel of the road to the place of performed tasks, as well as after their completion.



Fig. 5. Measuring elements during installation in a mobile devices

The preliminary analysis on selected objects show high variability of ecological parameters. Through the curves obtained, we can analyze the impact of various factors on emission changes relative to the change in speed and load resulting from the specifics of the device's operation. The following characteristic (Fig. 6) shows how important the test conditions are for the selected example for a research object. Visible changes in HC emissions show how reasonable it is to strive for tests under real operating conditions. They allow showing real values during operation. Differences resulting from work between static conditions and test dynamic conditions reach a discrepancy at the level of growth reaching 25%. The application of the presented research methodology also gives new possibilities for non-road mobile devices.

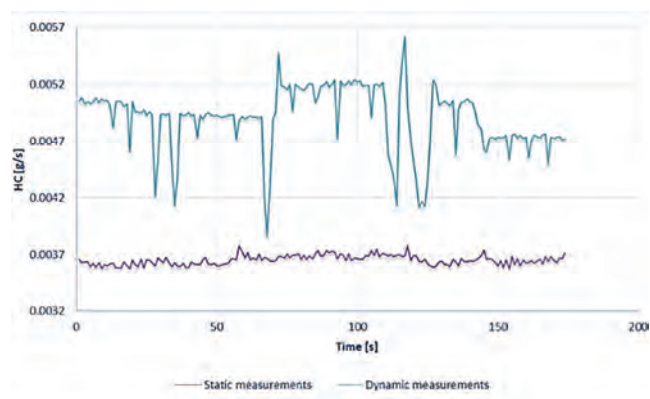


Fig. 6. The dependence of HC [g/s] emission under test conditions

It also provides information about real fuel consumption using the carbon balance method according to the following formula:

$$FC_w = \frac{0.1155}{\rho_{fuel}} [(0.866 \cdot HC) + (0.429 \cdot CO) + (0.273 \cdot CO_2)] \quad (1)$$

where: FC – fuel consumption [dm³/100 km]; HC, CO, CO₂ – emission of ingredients harmful [g/km]; ρ_{fuel} – fuel density at temperature 15°C [g/cm³].

Nomenclature

CI	compression ignition	PM	particulate matter
GPS	global positioning system	RDE	real driving emissions
NDIR	non-dispersive infrared	SI	spark ignition
PEMS	portable emission measurement system	T	temperature

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Due to significant construction constraints (the impossibility of assembling of measuring elements) up to this time the indications were usually limited to organoleptic control.

4. Summary

The problem of non-road mobile machines presented in this article is highly significant. The currently produced drive units for this category of equipment are not subject to legal regulations and emission standards such as those required for approval tests. It causes their significant deviations from any levels for emission limits in relation to commonly used drive units mounted even in vehicles. The undertaken works indicate, that the load and speed of the engine have a significant impact on the exhaust emissions of the tested engines. The operation of the engines in small devices is characterized by changes in engine speed and engine load that are likely to cause relatively high exhaust emissions. Initial analyzes confirm that the direction of the undertaken activities will enable a more precise definition of the problem that is the negative environmental impact of the operated equipment. The creation of a universal methodology in research during real working conditions can be successfully used in the development of small engines installed in devices dedicated to manual work. It is necessary however, to continue development work on improving not only the measuring equipment intended for the engine of these devices, engines themselves, but also emission assessment methods. Measurements carried out on small motors under static conditions differ from measurements made under the conditions of the final use. An important issue is the adaptation of PEMS measuring equipment. Most of these devices have exhaust system components and a mass flow range beyond the range corresponding to the small engine category. An additional problem is the need for a compact design of measuring equipment. Carefully analyzing current trends in engine exhaust emissions, seems it is entirely reasonable to undertake emission tests related to small engines operating under real operating conditions. In the future, such tests should be included in the approval procedure.

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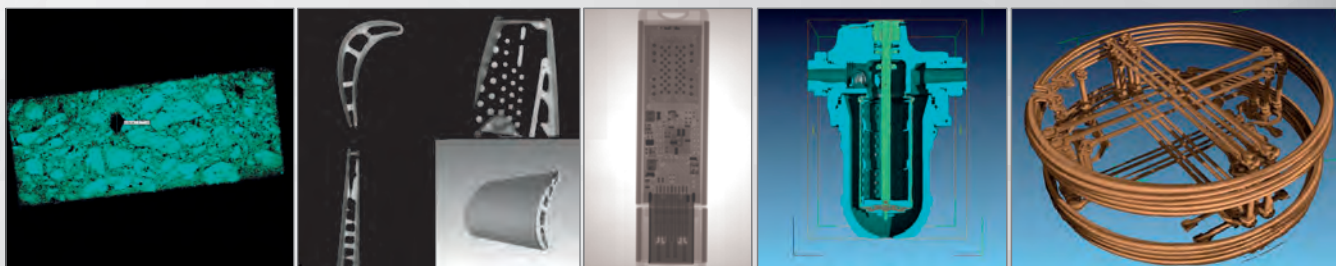


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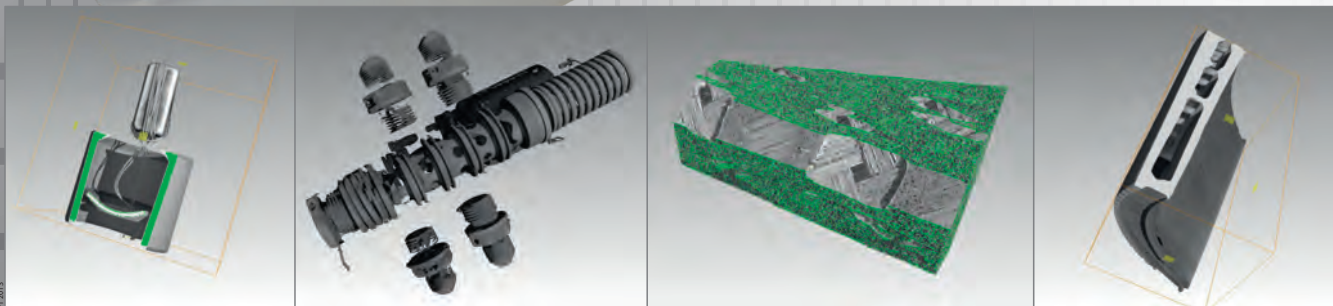
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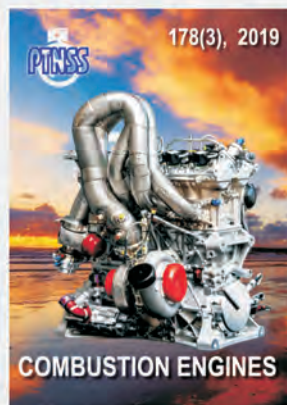
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