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ANALYSIS OF THE SHARE OF NATURAL GAS IN THE TOTAL FUEL SUPPLY DOSE ON THE COMBUSTION PROCESS IN A CRDI ENGINE

Summary. Natural gas is one of the potential combustion engine fuels whose proportion in the overall energy balance is expected to increase. Owing to some of its properties, its use requires a dual-fuel supply system; thus, the use of natural gas as a fuel for diesel engines is currently limited. Systems that supply gas fuel to diesel engines do not usually interfere with the engine control system. This solution significantly reduces system-installation costs. However, as demonstrated in the present study, it considerably changes the course of the combustion process, which increases thermal and mechanical loads. In this case, the combustion process can be controlled by changing the liquid fuel injection pressure or advancing the injection angle. This, however, requires interference with the engine control system.

1. INTRODUCTION

Numerous efforts have been made to reduce toxic compound emissions into the atmosphere by means of transport, and limited oil resources necessitate the search for new combustion engine fuels. When analysing the current trend of research into the use of alternative combustion engine fuels, it should be assumed that the significance of gaseous fuels will continue to increase in the near future. Natural gas will represent the largest proportion of gaseous fuels due to its relatively large resources and its favourable hydrogen-to-carbon ratio, which guarantees reduced CO_2 emissions [1-5].

The main component of natural gas is methane. Therefore, this fuel can be used directly to power spark-ignition (SI) engines. The use of natural gas is much more efficient than diesel, as diesel engines are hampered due to the high methane compression ignition (CI) temperature (approx. 840 K), which necessitates the use of an external ignition source [2,6]. The most commonly used source of gaseous fuel ignition in diesel engines is a small dose of diesel fuel, whose CI triggers the combustion of gas and air mixture. Currently, the main reason for using natural gas to power CI engines is to reduce operating costs. Commercially available systems for engines of this type enable a relatively simple adaptation of the engines to operate within a dual-fuel system. Unfortunately, they are primarily used to reduce fuel consumption costs; much less attention is given to optimising the combustion process, which has a major impact on the overall engine efficiency and toxic compound emissions [6-7].

In engines operating on liquid fuel, the course of combustion can be controlled by modifying the fuel injection characteristics *inter alia* by dividing the fuel dose into several portions or changing the injection pressure. Regarding the diesel engine dual-fuel supply, natural gas is generally supplied with

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air as the cylinder is filled with a homogeneous gas and air mixture with a constant excess air ratio. The injection of the first liquid-fuel dose can trigger the combustion of gaseous fuel found in the cylinder, and further fuel propagation is determined by the mixture parameters. Therefore, in this case, the control of the combustion course is limited to controlling the ignition onset by selecting the liquid fuel dose, its spraying, and the injection advance angle. Further gaseous fuel combustion is specified by both the conditions prevailing in the cylinder and the properties of the gaseous fuel and air [8-12].

Therefore, the course of combustion in a dual-fuel CI can be expected to differ considerably from the course of combustion in a classical engine, which has a considerable effect on engine efficiency and the levels of toxic compound emissions into the atmosphere.

Several previous studies [2, 6-7, 10-13] involved the use of a fuel supply system programmed to operate with liquid fuel was used. The results indicate that increasing the proportion of gaseous fuel in the supply dose reduces the temperature during compression, which prolongs the fuel self-ignition lag. Meanwhile, in a later combustion phase, the heat release rate increases, as do the maximum combustion temperatures. The results clearly show that for diesel engine dual-fuel supply, the way of fuel supply control must be different from the single-fuel operation.

The course of combustion in a dual-fuel engine can also be controlled by changing the pilot dose injection pressure [8, 14]. A change in the injection pressure allows the pilot dose injection time to be changed. It also enables control over the range of the liquid fuel jet sprayed in the cylinder, which affects the flame propagation in the cylinder. Another way to control the course of combustion in a dual-fuel engine is to adjust the pilot dose injection advance angle [11, 15-17].

Therefore, for engine dual-fuel supply, new control systems need to be developed to optimise engine operation. To this end, numerous research centres are carrying investigating the effect of individual control parameters. The vast majority of these studies concern the effect of individual parameters on toxic compound emissions [1, 11-13, 18-20].

The cited literature has focused primarily on the environmental aspects of the use of gaseous fuels in diesel engines. It can be assumed, however, that a change in the course of combustion, which results in a change in the pressure increase acceleration, can alter the mechanical loads, while a change in the course of heat evolution can result in a change in thermal loads on the engine. The vast majority of studies conducted to date contain no reference to the contribution of such a supply to a change in mechanical and thermal loads that significantly influence engine operation.

2. METHODOLOGY AND THE STUDY OBJECT

Tests for the possible use of NG to fuel CI engines were conducted on an ADCR-type engine (Andoria Common Rail). The technical specifications of this engine are shown in Table 1 [3].

The test engine was mounted on an engine test bench (Fig. 1) comprised of:

- an eddy current brake AVL DynoPerform 240;
- a fuel consumption measurement system AVL 735S;
- an engine speed control system AVL THA100;
- an engine test bench management system; and
- a PUMA Open measurement data acquisition system.

The dual-fuel supply of the engine was enabled by installing a gas mixer upstream of the compressor in the inlet system.

During the tests, the engine was supplied with fuel in two ways:

- using a manufactured Bosch Common Rail 2.0 fuel supply system controlled by a Bosch EDC16C39 controller and
- using a laboratory supply system that enabled the adjustment of the dose, pressure, and liquid fuel injection advance angle values.

The applied measurement system ensured the measurement of basic engine operating parameters (i.e. fuel consumption, torque, and engine speed). In addition, the engine test bench control board allowed the pre-determined constant crankshaft speed to be maintained. It made it possible to maintain the engine load at the pre-set level due to the coupling of the engine load controller with the engine test bench control system [21].

Table 1

Technical specifications of the ADCR engine [3]

Engine	**ADCR**
Engine type	4-stroke diesel engine, turbocharged
Fuel injection	Common Rail 2.0
Cylinder number and arrangement	4 cylinders
Swept volume	2636 cm^3
Compression ratio	17.5 : 1
Cylinder diameter/piston stroke	94/95 mm
Maximum torque/rotational speed	250 N·m/1800-2200 rpm
Power rating / rotational speed (according to ISO 1585)	85 kW / 3700 rpm
Specific fuel consumption at the maximum engine torque (according to ISO 1585)	210 g/kWh
Minimum idle speed	750 rpm
Turbosupercharger	radial, with an exhaust gas vent valve
Exhaust gas recirculation	EGR pneumatic

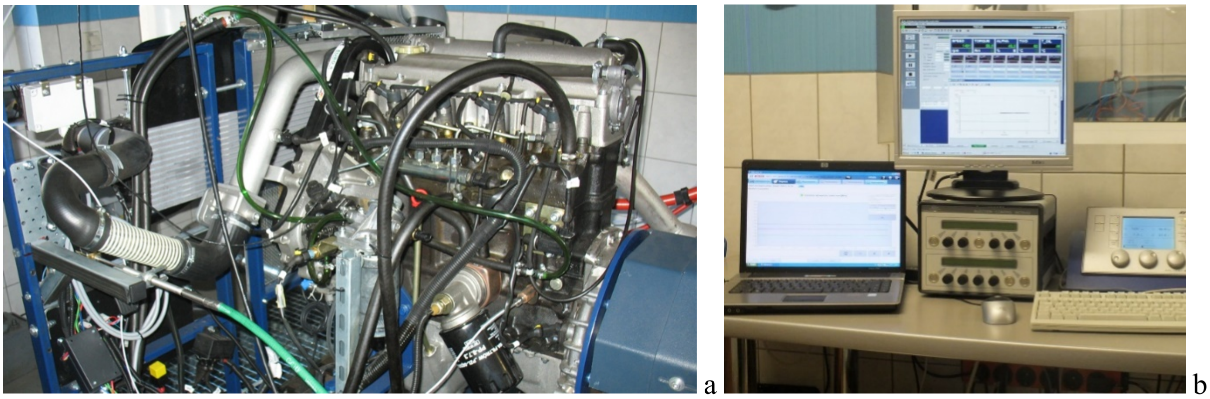

Fig. 1. (a) The test engine and (b) the engine test bench control board

A Kistler 6056A sensor adapter was screwed in instead of the heating plug in the first engine cylinder to record pressure changes in the combustion chamber [3]. The measurement of the instantaneous position of the engine crankshaft was ensured by installing an encoder on the free end of the crankshaft. In this case, the angle marker resolution was 720 points per revolution. E-type high-methane natural gas (available in the municipal grid) with a methane content of approximately 97.8% and a declared calorific value of 36,335-36,374 MJ/m^3 was used as the gaseous fuel.

The pressure courses in the combustion chamber as a function of the crank angle (recorded during the measurements) were used to calculate the combustion parameters. The combustion course was analysed based on 50 successive engine operating cycles. Before the recorded results were used for calculations, the recorded courses were averaged by calculating the arithmetic mean from the recorded courses for each course point. Numerical procedures were applied for further calculations. Precise descriptions of the methods were presented by Shepel et al. [3].

3. STUDY RESULTS AND THEIR ANALYSIS

3.1. The effect of natural gas on the course of combustion

As previously mentioned, during the first stage of testing, the engine was supplied using a conventional CR 2.0. The engine operation was controlled by an EDC16C39 controller programmed to operate on liquid fuel. The air supply system was modified to supply gaseous fuel to the engine, and a gaseous fuel supply system was mounted upstream of the turbosupercharger. This solution supplied a homogeneous mixture of gaseous fuel and air to the engine.

The energy contribution of gaseous fuel (U_g) in the engine supply dose was determined based on the following equation:

$$U_g = \frac{\dot{m}_{NG} \cdot W_{NG}}{\dot{m}_{df} \cdot W_{df}} \cdot 100\% \tag{1}$$

where:
$\dot{m}_{df}$ - diesel oil jet during the single-fuel operation [kg/min];
$\dot{m}_{NG}$ - CNG jet supplied to the engine [Nl/min];
W_{df}- diesel oil calorific value [kJ/kg]; and
W_{NG} - NG calorific value [kJ/Nl].

Therefore, the percentage of the energy supplied with gaseous fuel U_g indicated the amount of energy supplied to the engine in the form of gaseous fuel in relation to the energy supplied to the engine with liquid fuel during the single-fuel operation.

The controller, which was initially applied to control the test engine depending on the load, followed two fuel injection strategies. At lower loads, the fuel dose was divided into two portions – the initial jet and the main dose – while at higher loads, a single fuel dose was injected.

Fig. 2 shows the course of pressure changes in the test engine cylinder and the calculated engine operating parameters at a rotational speed of 1500 rpm and a load of 100 Nm during the operation of the engine supplied with diesel oil and during the operation of the engine in a dual-fuel system with 50 and 75% proportions of gaseous fuel.

The pressure courses show that increasing the gaseous fuel proportion in the supply dose increases the maximum pressure in the combustion chamber increases. An analysis of the heat evolution course shows that an increase in the fuel proportion in the supply dose accelerates the heat release. This is because, for the analysed loads, a diesel oil pilot dose is injected, which burns under the single-fuel operation conditions and improves the main dose combustion conditions. Meanwhile, when natural gas is present in the cylinder, the pilot dose self-ignition triggers the combustion of NG, whose combustion intensity is determined by the properties of the gas and air mixture. At high gaseous fuel proportions, the self-ignition of the pilot dose triggers gaseous fuel combustion within the entire combustion chamber, which considerably accelerates heat release and reduces overall engine efficiency. The earlier heat release results in a more rapid temperature increase in the combustion chamber, which (at large gaseous fuel proportions) increases the maximum temperature. An analysis of the course of pressure increment changes in the cylinder shows that, since the pressure increment does not exceed 0.2 MPa/crankshaft rotation degree, no excessive loads are imposed on the piston-crank system.

The testing results at a speed n=3000 rpm and a load of 100 Nm, obtained for the engine operating under the single-fuel operation conditions at 45% and 65% proportions of gaseous fuel, are provided in Fig. 3. At this rotational speed, the fuel dose is injected as a single dose at high loads and is divided into two portions at low loads. In the single-fuel operation in the case being analysed, one fuel dose is injected, similarly to the option of the 45% proportion of gaseous fuel. At gaseous fuel proportions exceeding 50%, the fuel dose is divided into two portions due to the considerable reduction in the diesel fuel dose.

The results presented in Fig. 3 show that higher engine loads and significant proportions of gaseous fuel fundamentally change the course of combustion. During the load compression in the cylinder, the temperature decreases considerably, which prolongs the self-ignition lag. However, for small doses of diesel fuel – meaning that the fuel dose is divided into two portions –the early injection of the liquid fuel pilot dose results in the premature ignition of gaseous fuel, which leads to earlier heat release, contributing to a reduction in engine efficiency. At significant gaseous fuel proportions in the supply dose, the pressure increment rate (*dp/dα*) also increases, which can increase the dynamic loads imposed on the piston-crank system and the power transmission system.

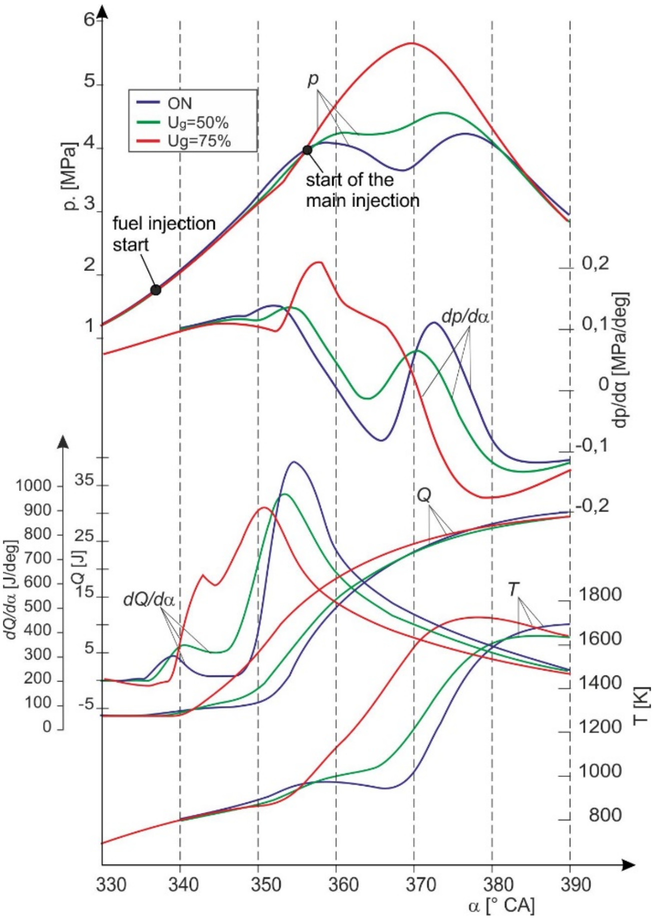

Fig. 2. The course of changes in combustion parameters depending on the share of natural gas in the supply dose at n=1500 and M= 100 Nm

Fig. 4 shows the effect of the natural gas proportion in the engine supply dose on the overall efficiency of an engine using a system programmed for the standard operation with liquid fuel. The figure clearly shows that an increase in the gaseous fuel proportion in the supply dose, where this engine supply method is applied, decreases the overall engine efficiency. This is primarily due to the change in the course of combustion in the cylinder, which is caused by the presence of natural gas. The decrease in overall engine efficiency increases the thermal loads imposed on the engine.

3.2. The effect of liquid fuel pilot dose parameters on the course of combustion

The conventional solution of the common rail system fails to ensure the proper progress of the combustion process and causes a significant reduction in engine efficiency. Therefore, tests were conducted to determine the effect of individual liquid fuel dose parameters on the course of the gaseous fuel combustion process. To this end, the conventional supply system was replaced with a laboratory system, which the diesel oil pilot dose parameters to be changed.

Fig. 5 shows the effect of the liquid fuel pilot dose injection pressure on the pressure course, as well as the release of heat in the combustion chamber at a 20% diesel oil dose. Fig. 6 shows the changes in the overall engine efficiency due to the pressure of the initial fuel dose.

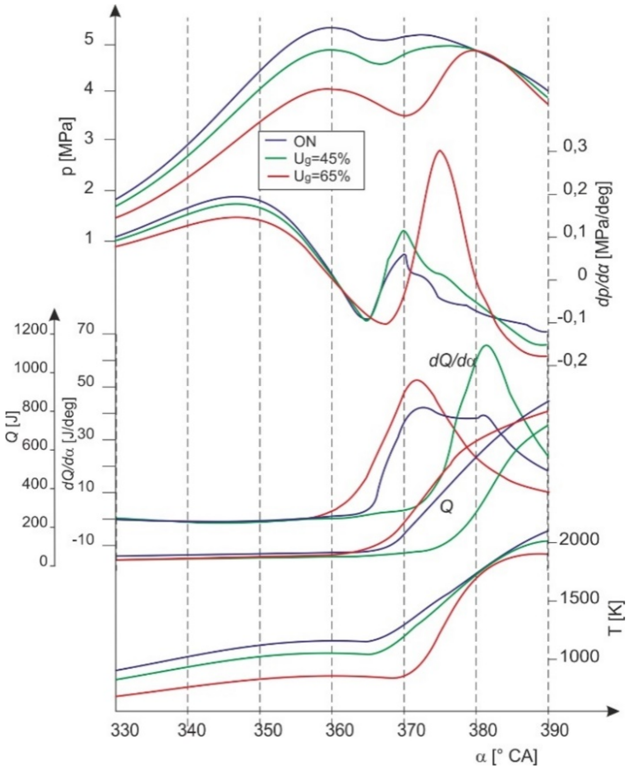

Fig. 3. The course of changes in combustion parameters depending on the share of natural gas in the supply dose at n=3000 and M= 100 Nm

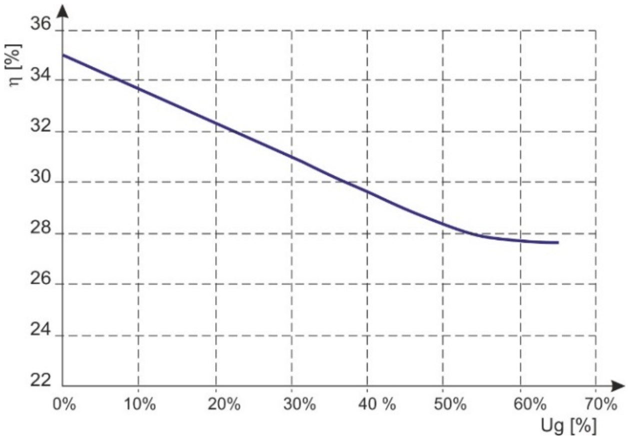

Fig. 4. Differences in the overall engine efficiency based on the share of natural gas in the fuel supply dose at n=3000 rpm and M=200 Nm

As Fig. 5 shows, the diesel oil pilot dose injection pressure significantly changes in the course of combustion. Along with an increase in the injection pressure of the liquid fuel dose, the range of the sprayed fuel jet increases, which improves flame propagation in the cylinder. This speeds up heat release and increases the temperature in the engine cylinder. Increasing the liquid fuel pilot dose injection pressure enhances engine efficiency beyond the efficiency achieved with the single-fuel operation. Unfortunately, increasing the pilot dose injection pressure also significantly increases the pressure in the cylinder and the pressure increments as a function of the crankshaft rotations. These outcomes result in higher mechanical loads on the piston-crank system.

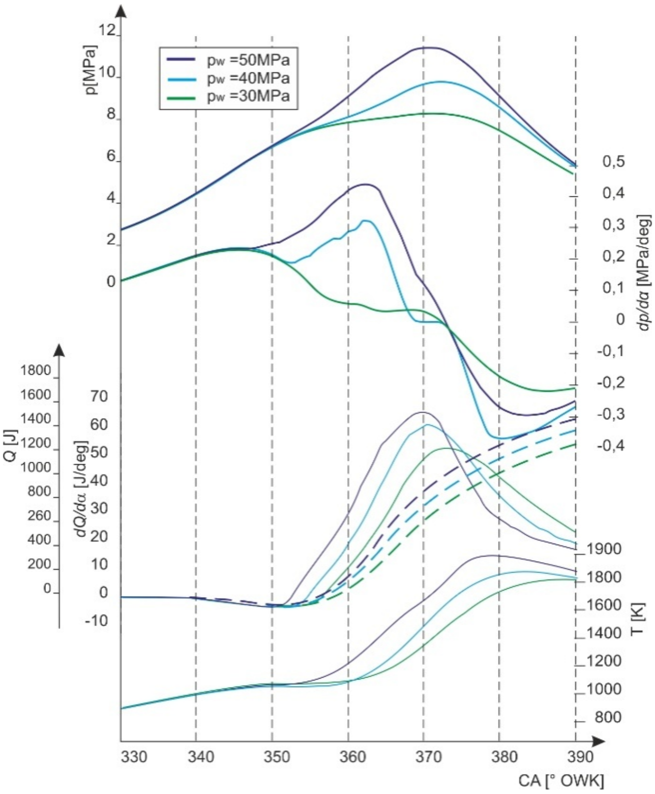

Fig. 5. The course of changes in combustion parameters depending on the injection pressure at n=3000 with a 20% liquid fuel supply dose

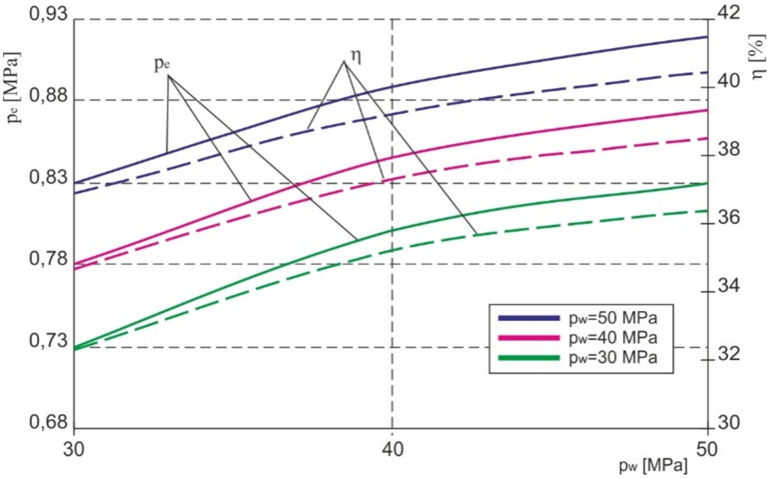

Fig. 6. Influence of the injection pressure of the constant fuel dose on the value of effective pressure and the overall efficiency of the engine at n=3000 rpm with a 20% liquid fuel supply dose

The effect of the initial dose of diesel fuel oil injection advance angle on the course of natural gas combustion in the combustion chamber is presented in Fig. 7 (with a diesel oil dose of 20%). The figure clearly shows that the injection advance angle has a substantial effect on the course of pressure increment and heat release. Increasing the initial fuel dose injection angle causes the combustion of gas and air mixture to occur earlier, which considerably increases the pressure in the engine cylinder and accelerates heat release.

The effect of the initial liquid fuel dose injection advance angle on the course of changes in effective pressure and the overall engine efficiency is presented in Fig. 8. The figure clearly shows that the injection advance significantly affects engine efficiency. However, at higher angle values, the maximum pressure in the engine cylinder increases considerably (Fig. 7), which increases the load imposed on the piston-crank system.

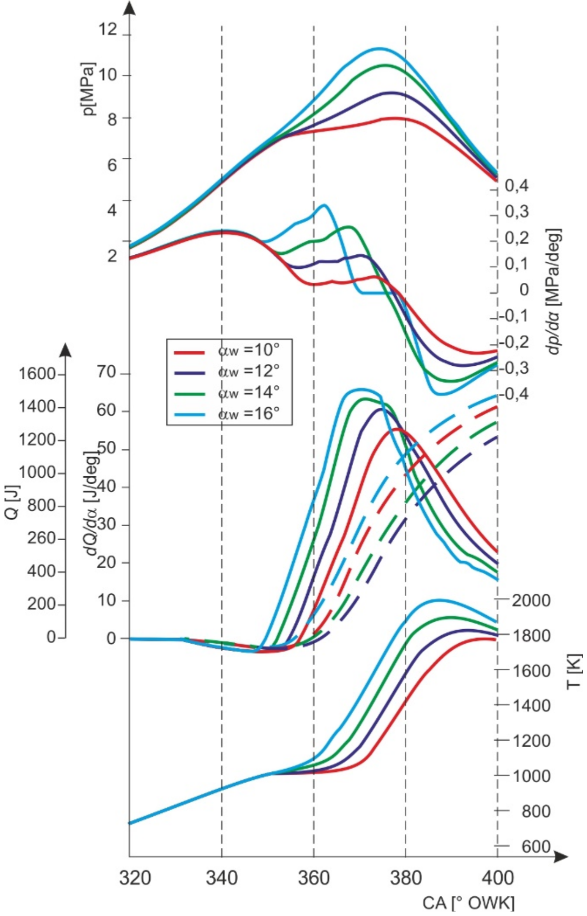

Fig. 7. The course of changes in combustion parameters depending on the injection advance angle at n=3000 with a 20% liquid fuel supply dose

4. CONCLUSIONS

The results of the study on the use of natural gas to power compression ignition engines led to the following conclusions:

1. The gaseous fuel proportion in the supply dose of a compression ignition engine has a significant effect on the course of the combustion process, which, in turn, determines the overall engine efficiency and the mechanical and thermal loads imposed on the engine. For relatively small gaseous fuel proportions (i.e. less than 50% of the total supply dose), the combustion process differs slightly from the combustion process in single-fuel operation. Meanwhile, greater proportions of gaseous fuel in the supply dose result in excessive pressure increases in the combustion chamber, which considerably increases the mechanical and thermal loads imposed on the engine.
2. In modern common rail fuel supply systems, the premature injection of the diesel oil pilot dose can trigger the combustion of natural gas, which adversely affects the course of combustion while generating excessive loads on the piston-crank system and the engine cooling system. An excessively large proportion of gaseous fuel in the supply dose can result in an uncontrolled course of combustion.
3. The course of combustion in a dual-fuel compression ignition engine can be controlled by appropriately selecting the liquid injection pressure and changing the diesel oil injection advance angle. However, these tasks interfere with the engine control system, which considerably increases

the cost of installing the system supplying liquid fuel to the engine while allowing the combustion in the engine to be controlled. This outcome increases engine efficiency and lightens mechanical and thermal loads imposed on individual engine systems.

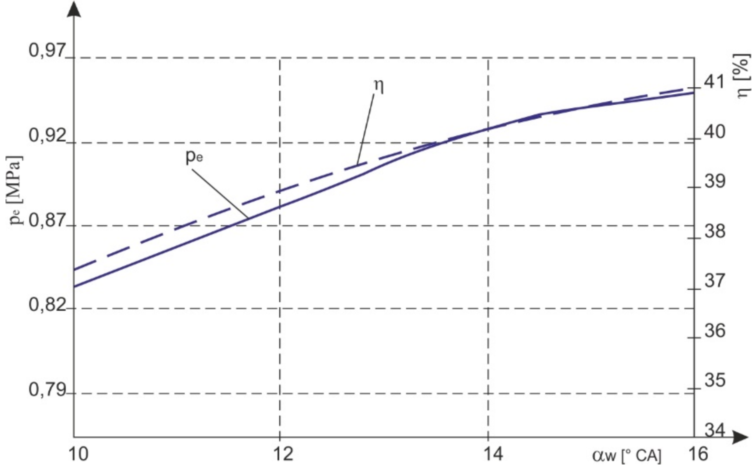

Fig. 8. The effect of pilot dose injection advance angle on the effective pressure value and the overall engine efficiency at n=3000 rpm

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