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THE INFLUENCE OF THE INJECTION TIMING ON THE PERFORMANCE OF TWO-STROKE OPPOSED-PISTON DIESEL ENGINE

Abstract

The performance of the engine strongly depends on the parameters of the combustion process. In compression ignition engines, the fuel injection timing has a significant influence on this process. The moment of its occurrence and its duration should be chosen so that the maximum pressure value occurs several degrees after TDC. In order to analyze the effect of the fuel injection timing on the performance of the tested two-stroke opposed-piston diesel engine, a zero-dimensional model was developed in the AVL BOOST program. Next, a series of simulations were performed based on the defined calculation points for maximum continuous power, which resulted in power, specific fuel consumption and mean in-cylinder pressure. Finally, the engine map was made as a function of the start of combustion angle.

1. INTRODUCTION

Two-stroke opposed piston engines have a number of advantages over four-stroke engines. Due to the lack of a timing system, there are less thermal losses. The lack of timing system translates into greater reliability, cheaper operation and simpler service. In addition, in this type of engines one stroke per revolution of the crankshaft is produced, and thus more power is obtained compared to a four-stroke engine with the same capacity. The advantage of opposed piston engines is good balance, the possibility of using different fuels and simple

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injectors (Pirault, 2010). Regner et al. (2011) and Regner, Koszewnik, and Venugopal (2014) presented a number of advantages of the proposed two-stroke opposed piston diesel engine. However, four-stroke engines are better in terms of the emission of toxic substances as well as the heat load of the components of the crankshaft and the wear of the cylinder liner and the rings.

So far, these types of engines have been used on ships, in military vehicles and special machines. The wide range of high-quality catalysts and lubricants currently available on the market creates new opportunities for the use of two-stroke opposed piston engines, especially in land and air transport.

Many tests of this type of engines have been performed. An example is work written by Ma et al. (2015), in which the combustion and heat dissipation process in an opposed piston diesel engine was studied. Abani et al. (2017) presented a number of assumptions and design solutions applied in the design of a two-stroke opposed piston diesel engine that achieves 55% thermal efficiency. Another example is the study of a two-stroke opposed piston diesel engine for propulsion of light aircraft performed in the work written by Cantore, Mattarelli, and Rinaldini (2014).

In self-ignition engines, the injection advance angle is a parameter significantly affecting the process of formation of a fuel-air mixture (propagation of fuel drops after combustion, evaporation and mixing with air) and the combustion process (speed and amount of heat produced). Its value depends on the moment when the pressure in the cylinder increases dramatically in relation to the piston position in the TDC. In order to obtain the highest efficiency, this angle should be selected so that the maximum pressure occurs several degrees CA after the piston TDC.

The injection advance angle is clearly linked to the angle of the start of combustion. An earlier fuel injection results in an earlier formation of a fuel-air mixture, and therefore an earlier initiation of the combustion process. In the work written by Katarasnik, Trenc, and Opresnik (2006), the criterion for determining the start of combustion based on the analysis of the third derivative of the mean pressure signal in the cylinder as a function of the angle of rotation of the crankshaft has been presented.

Agarwal et al. (2013) conducted experimental studies of a single-cylinder diesel engine, in which the impact of injection advance and injection pressure on selected engine performance parameters (including cylinder pressure, rate of heat release, specific fuel consumption, exhaust emissions) were analyzed.

In recent years, many works have been published devoted to the study of dual-fueled engines. The examples are works written by Sayin, and Canakci (2009) and Sayin, Ilhan, Canakci, and Gumus (2009), in which the influence of the injection advance angle on the performance of a single-cylinder diesel engine fueled by a mixture of diesel with ethanol and diesel with methanol, respectively. The tests have shown that increasing the injection advance angle in the engine running on the test mixture significantly reduces the emission of toxic substances (carbon monoxide and unburnt hydrocarbons).

Research was also performed on the influence of ignition timing on the performance of CI engines fueled by unconventional types of oils: used edible oil (Bari, Yu & Lim, 2004) plastic oil (Mani & Nagarajan, 2009) and bio-diesel (Ganapathy, Gakkhar & Murugesan, 2011).

Another important parameter correlated with the angle of advance of injection is the ignition delay. This is the period between the moment the fuel appears on the injector and the first self-ignition spots appear (Heywood, 1988). There are many studies devoted to the study of this parameter in diesel engines.

An example is work written by Assanis, Filipi, Fiveland, and Syrimis (2003), which shows the correlation of the ignition delay with physical and chemical processes based on pressure, temperature and stoichiometry.

Research performed by Kobori, Kamimoto, and Aradi (2000) showed that the ignition delay can be shortened by increasing the temperature and pressure inside the cylinder. In addition, increasing the injection pressure, the number of cetane fuel or reducing the diameter of the hole in the injector results in a shorter ignition delay.

The change of fuel type affects the ignition delay, as shown in the research performed in the publication written by Saho, and Das (2009). The study investigated the use of selected vegetable oils as a fuel in diesel engines and their effect on the combustion process.

As part of this work, a model of a two-stroke opposed piston diesel engine was presented. Based on the developed model, simulation calculations were made to investigate the effect of the injection advance angle on the performance of the designed opposed piston diesel engine.

2. ENGINE MODEL

In order to perform simulation tests on the influence of the injection advance angle on the performance of the designed engine, a zero-dimensional model was developed using the AVL BOOST program (Fig. 1).

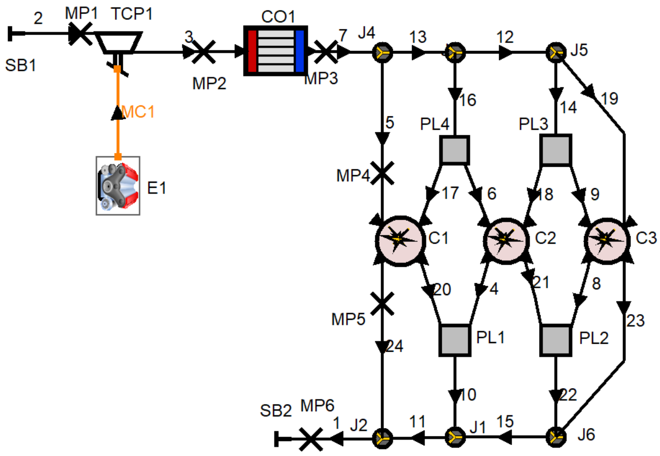


Fig. 1. Model of tested engine created in the AVL BOOST program

In the zero-dimensional model, the flame propagation method is not included, and all thermodynamic parameters are averaged in the working volume of the cylinder. This model is based on the first law of thermodynamics. The independent variable is the time, and the mass burning rate is assumed based on literature data (Heywood, 1988).

Figure 2 shows the basic settings for the calculation cycle defined in the AVL BOOST program.

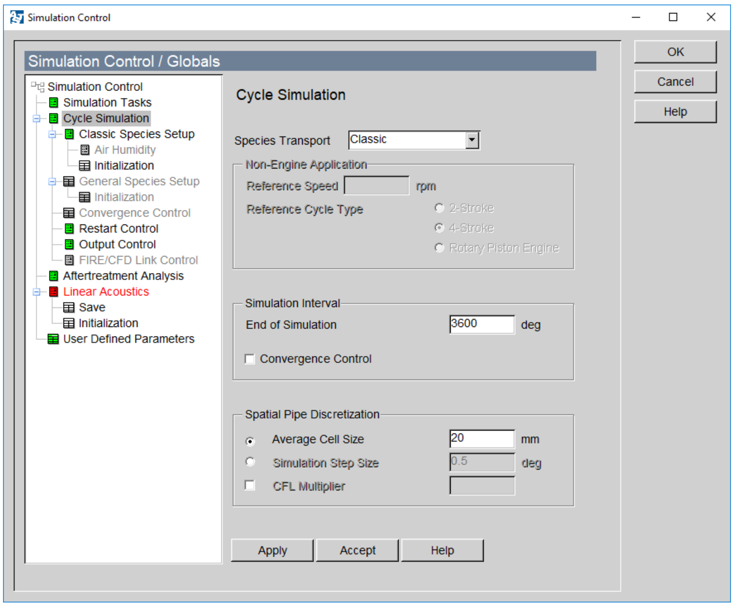


Fig. 2. Basic settings of the calculation cycle defined in AVL BOOST program

The Viebe function was used to describe the combustion process, which describes the rate of heat release (*ROHR*) as a function of the crank angle. After its integration, a part of the burnt mass (mass fraction burned, *MFB*) is obtained as a function of the crank angle. The Viebe function depends on four basic parameters: start of combustion (*SoC*), combustion duration (*CD*), curve shape parameter m and parameter a, depending on the burned part of the fuel injected (Stiesch, 2003). This function is the ratio of the heat released after a given angle of rotation of the crankshaft *Q(α)* and the total amount of heat released at the end of combustion Q_c. It has the form (1):

$$\frac{Q(\alpha)}{Q_c} = 1 - \exp\left(-a\left(\frac{\alpha - SoC}{CD}\right)^{m+1}\right) \tag{1}$$

where: *SoC* – start of combustion,
CD – combustion duration,
m – shape parameter,
a – parameter depending on the burned part of the fuel injected.

In the developed engine model, the following parameters of the combustion model were adopted:

- start of combustion *SoC*, from –10° to +10° relative to TDC, every 2°,
- combustion duration *CD* = 40°,
- curve shape parameter *m* = 0.7,
- parameter *a* = 6.9.

Fig. 3 shows the heat release coefficient (*ROHR*) and the mass fraction burned (*MFB*) as a function of the crank angle determined in the AVL BOOST program based on the defined parameters of the combustion model.

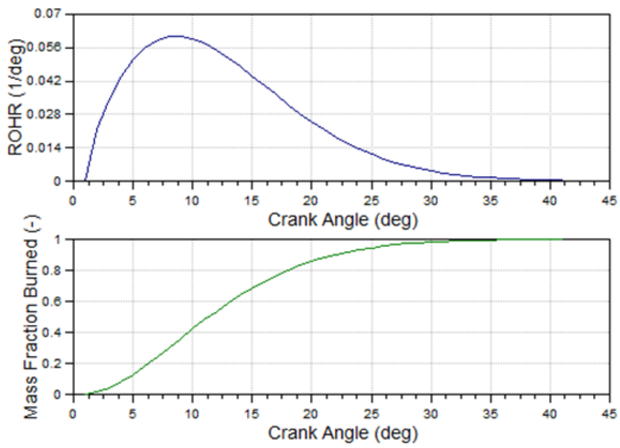


Fig. 3. ROHR and MFB as a function of the crank angle

In the model it was also necessary to define the model of heat exchange through the walls. The Woschni 1978 standard model was adopted, based on the Newton's law of cooling, in the form of (2):

$$\frac{dQ_w}{dt} = hA\left(T_w - T_g\right) \tag{2}$$

where: Q_w – heat flow from gas to walls,
h – heat exchange coefficient,
A – surface area,
T_w – average temperature of walls,
T_g – average gas temperature. (Stiesch, 2003)

In the heat exchange model, the surface area of the piston and cylinder was defined based on the design assumptions of the engine. In addition, based on data from work written by Pulkrabek (1997), temperatures were set on their surfaces. In order to take into account the effect of the load turbulence in the cylinder on the heat exchange coefficient *h*, the swirl ratio *SR* = 1.5 defined as the speed of the load rotation in relation to the engine speed was defined. Table 1 presents the basic technical parameters of a given engine.

Tab. 1. Basic technical parameters of a tested engine

Engine type	Two-stroke opposed piston diesel engine
Scavenging method	Mechanical compressor Rotrex C30-64 with intercooler
Cylinders numer	3
Piston bore d	65.5 mm
Piston stroke S	72 mm
Compression ratio ε	22:1
Maximum power N_{max}	100 kW (4200 rpm)
Offset O	14°

In order to improve the filling of the cylinders with a fresh charge in the two stroke opposed-piston engines, the difference in the working phases of the crankshafts is used. Crankshaft from the side of the inlet ports get ahead of the shaft from the side of outlet ports. This angle is called offset. For the tested engine model an offset equal to *O* = 14° was assumed.

The AVL BOOST program does not have the option of directly designing the opposed piston engine, therefore simplification has been applied, consisting in doubling the piston area, resulting in a calculated piston diameter of $d_{calc} = d\sqrt{2}$ (Fig. 4).

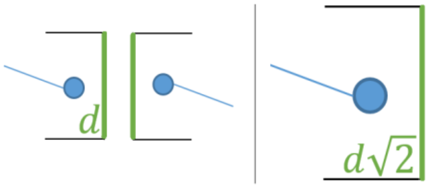


Fig. 4. The method of determining the calculated piston diameter

A number of geometrical parameters have been introduced to the model: the calculated diameter of the piston d_{calc}, the piston stroke S, the compression ratio ε, the length of the connecting rod l.

The model assumes a classic model of transport of chemical compounds (classic species transport). As a fuel, diesel oil with a calorific value of W_u = 42.8 kJ/kg and stoichiometric air-fuel ratio AFR = 14.7 kg of air per kg of fuel was adopted. The firing order was defined as follows: cylinder 1 – 0°, cylinder 2 – 120°, cylinder 3 – 240°. The engine model includes the friction model developed by Patton, Nitschke, and Heywood (1989).

The load was exchanged with a Rotrex C30-64 mechanical compressor driven mechanically from the engine shaft. Operating characteristics of the compressor was introduced to the model based on a map from the manufacturer ("Rotrex Technical Datasheet C30 Range", 2018). In addition, an air-to-air intercooler was used to improve the filling of the engine with a fresh charge behind the compressor. The scavenging model defined as scavenging efficiency as a function of the delivery ratio was introduced based on literature data (Blair, 1996). Figure 5 shows the defined scavenging model in cylinder settings window.

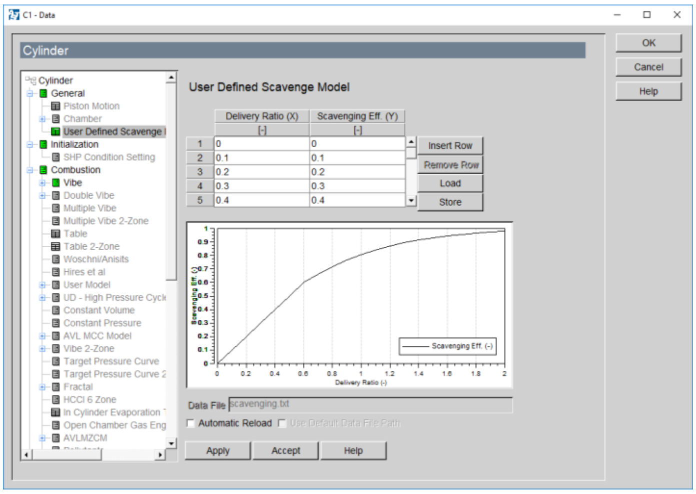
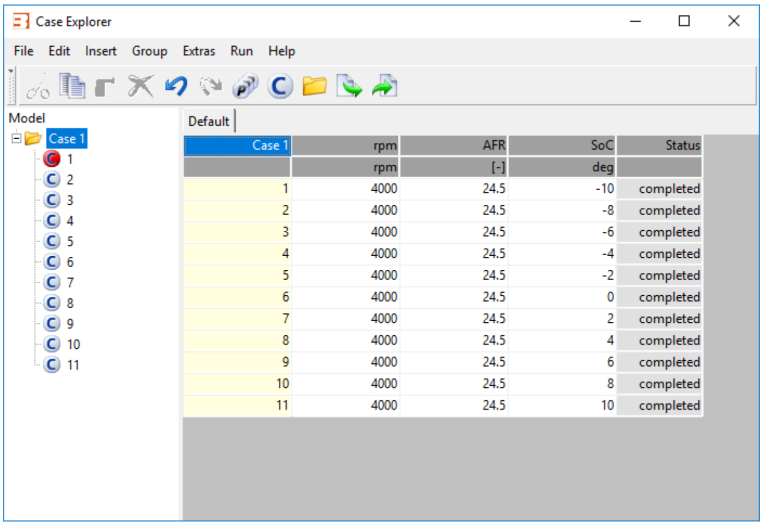


Fig. 5. Scavenging model defined in cylinder settings window in AVL BOOST program

In addition, inlet and outlet ports characteristics were introduced based on the engine design assumptions.

3. RESEARCH PLAN

During the tests, the influence of the injection timing on the performance of the analyzed engine (power, specific fuel consumption, mean cylinder pressure) was examined. The moment of the beginning of combustion depends on the injection timing, therefore the value of the angle at which the start of combustion (*SoC*) occurs is taken as the parameter. The tests were performed for a rotational speed of n = 4000 rpm and an air-fuel ratio equal to AFR = 24.5 kg air / kg fuel. The SoC parameter varied from –10 ° to +10 ° relative to TDC, every 2°. The defined measuring points are shown in Fig. 6.



Case 1	rpm	AFR	SoC	Status
	rpm	[-]	deg	
1	4000	24.5	-10	completed
2	4000	24.5	-8	completed
3	4000	24.5	-6	completed
4	4000	24.5	-4	completed
5	4000	24.5	-2	completed
6	4000	24.5	0	completed
7	4000	24.5	2	completed
8	4000	24.5	4	completed
9	4000	24.5	6	completed
10	4000	24.5	8	completed
11	4000	24.5	10	completed

Fig. 6. Defined measuring points in Case Explorer window in AVL BOOST program

4. RESULTS AND ANALYSIS

As a result of the performed calculations, the effective power N (Fig. 7), the value of specific fuel consumption g_e (Fig. 8) and the mean cylinder pressure as a function of crank angle (Fig. 9) were obtained for defined measurement points. The results of calculations are presented in Table 2.

Tab. 2. Calculation results

SoC (°)	*N* (kW)	g_e (g/kWh)	p_{max} (MPa)	$\alpha(p_{max})$ (°)
–10	84.85	222.05	17.62	5
–8	85.83	219.35	16.70	6
–6	86.31	217.91	15.77	7
–4	86.29	217.56	14.73	8
–2	85.52	218.48	13.65	9
0	85.19	219.74	12.62	11
2	83.85	222.49	11.54	12
4	82.41	226.06	10.54	14
6	80.45	230.66	9.54	15
8	78.64	235.82	9.39	0
10	76.66	241.85	9.39	0

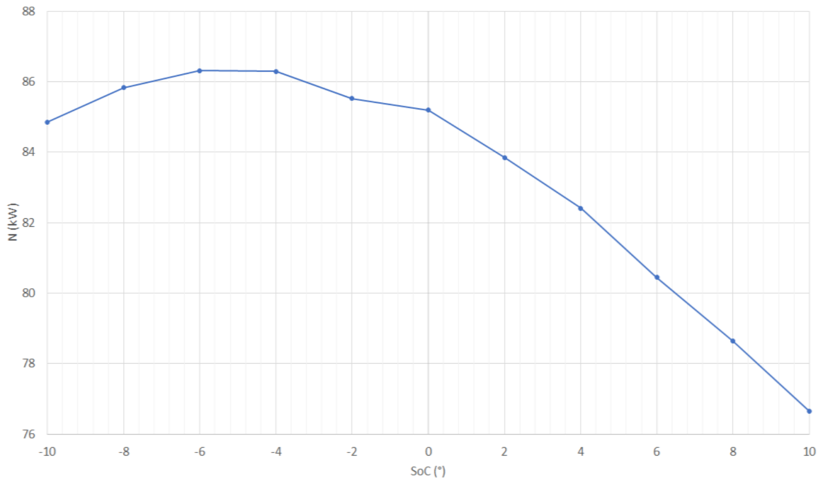


Fig. 7. Effective power as a function of SoC

The effective power has the highest value equal to 86.31 kW and 86.29 kW respectively for *SoC* = –6° and *SoC* = –4° angles. As the *SoC* angle increases or decreases, the effective power decreases. for an angle greater than *SoC* = 0°, the decrease is rapid.

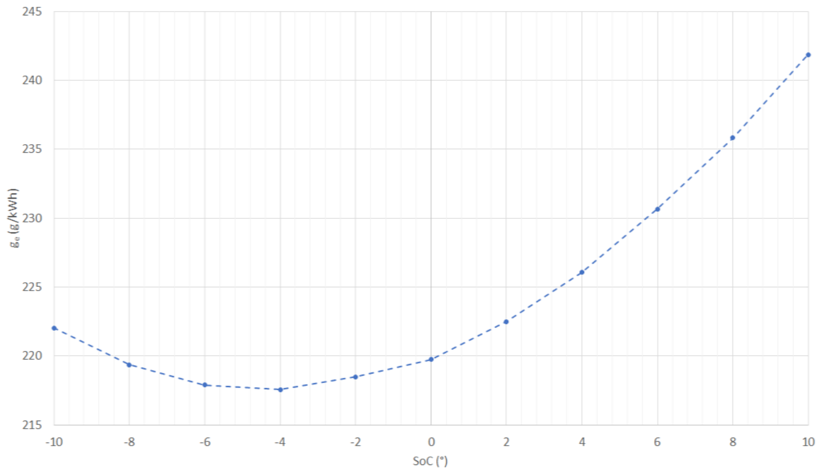


Fig. 8. Specific fuel consumption as a function of SoC [source: own study]

The specific fuel consumption has the smallest value equal to 217.91 g/kWh and 217.56 g/kWh for *SoC* = –6 ° and *SoC* = –4 ° respectively. With the increase or decrease of the *SoC* angle, the specific fuel consumption increases.

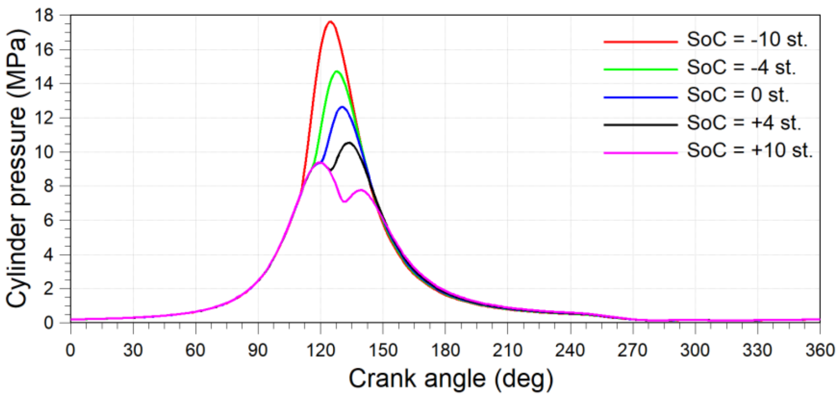


Fig. 9. The mean pressure in 2nd cylinder as a function of the crank angle for selected values of the SoC angle

As the value of the *SoC* angle decreases, the maximum pressure value p_{max} increases and the area under the plot increases (the indicated work increases). For positive values of the *SoC* angle, the indicated work decreases, whereas for the large values of this angle (10°) the second peak of pressure appears, which indicates an incorrect combustion process. The maximum pressure for the angle of *SoC* = –4° is for the crank angle *CA* = 128°, and therefore 8° for the TDC of the piston.

On the basis of the cylinder pressure as a function of crank angle, the maximum pressure value p_{max} and the angle of occurrence of the maximum pressure after the TDC of the piston $\alpha(p_{max})$ were read from the defined measurement points (Fig. 10). For *SoC* = 8° and *SoC* = 10°, the value of $\alpha(p_{max})$ was 0, because in these cases there were two pressure peaks, the highest of which corresponded to the compression pressure in the TDC of the piston.

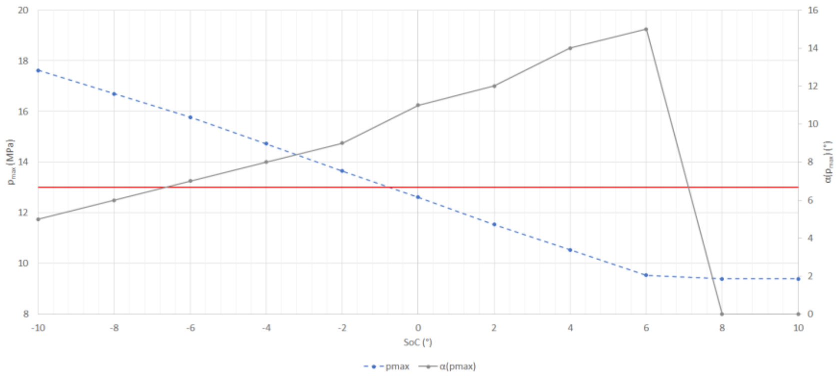


Fig. 10. Maximum cylinder pressure and the angle of occurrence of the maximum pressure after the TDC as a function of SoC angle

For reasons of strength of the crank and piston system, the maximum cylinder pressure should not exceed a certain limit value. Based on the design assumptions of the engine design, its value was assumed at 13 MPa. The value of the maximum pressure in the cylinder of the tested engine decreases with the increase of the value of the start of combustion angle (Fig. 10). It follows that the *SoC* angle for the engine under test should not be less than 1°, so as not to exceed the set maximum pressure limit in the cylinder. As the value of the *SoC* angle increases, the angle of occurrence of the maximum pressure after TDC of the piston $\alpha(p_{max})$ increases.

5. CONCLUSIONS

The fuel injection process depends on the injection timing and the combustion process. These processes have a significant impact on the engine performance. In addition, the injection timing translates into the moment of the start of combustion SoC, which determines the moment of rapid pressure increase in the cylinder. The calculations made it possible to analyze how the start of combustion SoC (and thus indirectly the injection timing) affects the effective power N, the specific fuel consumption g_e and the mean cylinder pressure p as a function of the crank angle. In addition, the limit value of *SoC* angle for the tested engine was specified.

For an optimum *SoC* angle value, the effective power generated reaches its maximum, while the specific fuel consumption assumes the minimum value. The optimum *SoC* angle for the designed engine ranges from –6° to –4°. For the *SoC* angle –4°, the effective power is approximately 86.29 kW, and the specific fuel consumption is 217.56 g/kWh. However, as the *SoC* angle decreases, the maximum pressure in the cylinder increases. This pressure should not exceed the permissible value resulting from the strength properties of the crank-piston system. It follows that the *SoC* angle for the designed engine should not be less than –1°.

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