

EFFECT OF VARIOUS IGNITION TIMINGS ON COMBUSTION PROCESS AND PERFORMANCE OF GASOLINE ENGINE

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Abstract

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This article deals with the effect of the ignition timing on the output parameters of a spark-ignition engine. The main assessed parameters include the output parameters of the engine (engine power and torque), cylinder pressure variation, heat generation and burn rate. However, the article also discusses the effect of the ignition timing on the temperature of exhaust gases, the indicated mean effective pressure, the combustion duration, combustion stability, etc. All measurements were performed in an engine test room in the Department of Technology and Automobile Transport at Mendel University in Brno, on a four-cylinder AUDI engine with a maximum power of 110 kW, as indicated by the manufacturer. To control and change the ignition timing of the engine, a fully programmable Magneti Marelli control unit was used. The experimental measurements were performed on 8 different ignition timings, from 18 °CA to 32 °CA BTDC at wide throttle open and a constant engine speed (2500 rpm), with a stoichiometric mixture fraction. The measurement results showed that as the ignition timing increases, the engine power and torque also increase. The increase in these parameters is a reflection of higher pressure in the cylinder, the maximum value of which is achieved at a higher ignition timing near top dead centre in the power stroke. In these conditions we can expect higher engine efficiency. It was also found that the combustion is more stable with a higher value of ignition timing. No significant difference was found in the combustion duration.

Keywords: spark timing, cylinder pressure, heat release, engine output

INTRODUCTION

Regulation has been associated with internal combustion engines since the start of their production. Thanks to the typical applications in which the engines were used, it was necessary to change their speed, or their power, depending on the requirements. For a very long time, regulation consisted of controlling the generation of the fuel mixture in the carburetor (quantitative control). It was necessary to realize, however, that the fuel through which its chemical energy is converted into mechanical work burns at a finite speed and has a difference course depending on ambient conditions. For this reason, it was also necessary to change the timing of the ignition of the mixture in

the cylinder, namely the ignition timing. For a very long time, controlling the ignition timing took place through a centrifugal and vacuum regulator located in the battery ignition manifold. (Martyr and Plint, 2007) Fuel prices have been growing in recent years, and tightening emission limits are forcing manufacturers to produce increasingly sophisticated internal combustion engines and their components, which will meet both legislative and user requirements. There are several ways to meet these requirements. The first way is using modern structural elements, such as high-pressure injection systems, fully electronic ignition systems, etc. (Mohan *et al.*, 2013) However, we could not use the potential of these systems today without good control elements. Mechanical control is

increasingly being replaced by electronic control – sensors, control units and actuators that can better use the potential of modern systems, depending on both the user's requirements and operating conditions. Operating conditions do not just include the speed, engine load and its temperature, but also the type of fuel used, especially recently, when biomass fuels, or biofuels, are being used. Thanks to the oxygen that alcohol-based biofuel (for spark-ignition engines) contains, they have a different octane number than gasoline (Bae and Kim, 2016). As a result of the higher octane number, and therefore better anti-knock properties, it is possible to adjust the ignition timing and achieve more favourable parameters. (Li *et al.*, 2010) For example, an earlier ignition timing at high loads reduces the exhaust temperature. For this reason, it is not necessary to enrich the mixture with fuel to protect the components from high temperatures. Reducing the injected dose will result in more economical operation of the engine. (Anderson *et al.*, 2012) Quality control of the ignition timing depending on operating conditions is reflected in lower fuel consumption, and it has an effect on the engine power and therefore its overall efficiency (Sayin, 2012). We can say that engine efficiency is a reflection of the quality of combustion. This is also confirmed by the fact that changing the ignition timing is reflected in the emissions - the amount of unburned hydrocarbons C_xH_y , nitrogen oxides NO_x and carbon oxides CO_x . (Binjuwair and Alkudsi, 2016)

This article evaluates the effect of the ignition timing on the basic parameters of a four-cylinder engine that runs on unleaded gasoline. The main assessed parameters include the output parameters of the engine, cylinder pressure variation, heat generation and burn rate. However, the article also discusses the effect of the ignition timing on the temperature of exhaust gas, the indicated

mean effective pressure, the combustion duration, combustion stability, etc.

MATERIALS AND METHODS

The measurements were performed in an engine test room in the Department of Technology and Automobile Transport at Mendel University in Brno. The tested engine was mounted on a test bench and connected to an eddy current dynamometer. The test was performed on a turbocharged, four-cylinder, spark-ignition four-stroke Audi engine. Other selected parameters of the engine are shown in Tab. I. The engine is equipped with a fully programmable Magneti Marelli engine control unit (ECU). The ECU, throttle and other equipment in the test room is controlled electronically from the control station in the test room, via a computer using LabVIEW software. No device for reducing the amount of pollutants in the exhaust gases was used during the measurement. The engine also did not drive common accessories (alternator, air conditioning compressor, power steering pump). For charge air cooling, an air/water intercooler was used instead of a serial air/air intercooler. The values of cylinder pressure were detected via special measuring spark plug with piezoelectric pressure sensor, which allows the cylinder pressure measurement without the effort of providing a special measuring bore. This sensor is mounted flush with the wall of the combustion chamber to keep its natural frequency at about 65 kHz. It is therefore also suitable for readings at high engine speeds and for knock control.

Experimental devices

Devices can be divided into three sections – measured engine, dynamometer for measurement of engine performance and apparatus for engine combustion analysis.

I: Selected parameters of measured engine (manufacturer's data)

Manufacturer	Audi
Type	APU
Maximal power [kW / HP]	110 / 150
Aspiration of the engine	turbocharger with intercooler
Intercooling	air/water
Maximal boost pressure [kPa]	167
Number of cylinders (disposition)	4 (inline engine)
Number of valves (per cylinder)	20 (5)
Displacement (per cylinder) [cm³]	1,781 (445)
Bore [mm]	81
Stroke [mm]	86.4
Conrod length [mm]	144
Compression ratio	9.5
Fuel	Natural 95
Maximum torque [Nm]	210
Cooling	fluid

- SI engine AUDI – mounted on a test bench and connected to the dynamometer,
- Electromagnetic eddy current dynamometer AVL DP 240 (see Tab. II),
- PC with LabVIEW software ,
- Kistler devices for engine combustion analysis:
 - measuring spark plug with piezoelectric cylinder pressure sensor type 6118BFD16, which is mounted in cylinder head instead of original spark plug, sensor is connected to KiBox,
 - crank angle adapter set 2619A – connected to inductive sensor on a crankshaft and also to KiBox,
 - system for combustion analysis KiBox To Go 2893AK1,
 - PC with software KiBoxCockpit – connected to KiBox via ethernet.

The measurement methodology is not determined by any standard or other regulation. The output engine parameter data are recorded through software created in LabVIEW. Indicators evaluating the combustion process, the high pressure engine indication, are read by a piezoelectric pressure sensor in the cylinder. The data are evaluated and recorded by measuring device KiBox through KiBoxCockpit software. The experiments were conducted at stoichiometric conditions for 8 different ignition timings, wide throttle open and a constant engine speed (see Tab. III). The detection of stoichiometric air-fuel mixture was performed via wide range oxygen sensor. The measuring spark plug with a pressure sensor was installed in the 1st cylinder instead of the original spark plug. The ignition timing is altered electronically in real time with an electronic control unit. The ECU receives values from various engine sensors - about the crankshaft position, the temperature and pressure in the intake manifold, the throttle angle, the composition of the fuel mixture through an oxygen sensor and the exhaust gas temperature. The turbocharged engine pressure sensor is situated between intercooler (immediately behind it) and throttle body (butterfly valve). Exhaust gas temperature sensing is performed by a thermocouple, which is situated in the exhaust manifold (in front of the turbocharger). Before the testing itself, the engine was brought up to operating temperature. For each measuring point

(change of ignition timing) a minimum period of two minutes was established for stabilization (stabilization of engine).

The indicator kit is equipped with a fully automated system of data collection which recorded an aggregate from 200 work cycles at a crank angle resolution 0.5 CA and averaged to calculate combustion-related parameters, with each work cycle being thermodynamically analysed, determining the basic combustion parameters and work process parameters. The aggregate thus allows for simple acquisition of the median values listed in this article. The thermodynamic calculation of the heat released is based on the first law of thermodynamics. The value is calculated indirectly from the cylinder pressure curve. The calculation works on the basis that the heat released in the cycle equals heat brought by the fuel reduced by the heat conducted into the walls of the cylinder. This allows a simplified calculation in real time. The calculation of the heat released operates in the range of -30°C to +90°C in relation to the piston top dead centre. After adjustments, we reach the following:

$$\frac{dQ}{d\alpha} = \frac{1}{\kappa - 1} \cdot \left[\kappa \cdot p \cdot \frac{dV}{d\alpha} + V \cdot \frac{dp}{d\alpha} \right] [J / ^\circ CA] \quad (1)$$

The data logger and software calculates the heat released in the evaluated element of volume change (calculation step) by the relation:

$$Q_i = \frac{konst.}{\kappa - 1} \cdot \left[\kappa \cdot p_i \cdot (V_{i+n} - V_{i-n}) + V_i \cdot (p_{i+n} - p_{i-n}) \right] [J] \quad (2)$$

where

Q ... heat released in the cycle [J],

V ... immediate volume above the piston [m³],

p ... absolute pressure in the cylinder [bar],

α ... crankshaft angle (CA) in relation to the piston top dead centre [°CA],

κ ... coefficient (depends on the specific heat capacity at constant volume c_v) [-],

i ... position of the evaluated element of volume change [°CA],

n ... size of the element of volume change (calculation step – 10°CA).

The calculated course of the heat input into circulation is used to determine important combustion parameters. Determining the beginning

II: Technical parameters of the dynamometer used

Manufacturer	AVL
Type	DP 240
Maximum power	$P_{max.} = 240 \text{ kW}$
Maximum torque	$M_{tmax.} = 600 \text{ Nm}$
Maximum revolutions	$n_{max.} = 10,000 \text{ rpm}$
Dynamometer regulator	speed and torque
Tensile force sensor	HOTTINGER U2 – 200 kg

III: Input measurement values

Engine speed [rpm]	2,500
Spark timing (ST) BTDC [°CA]	18, 20, 22, 24, 26, 28, 30, 32
Throttle [%]	100

and end of the combustion is considerably problematic. For this reason, the beginning and end of the combustion is determined by the crankshaft rotation degree relative to the top dead centre of the piston, when 5 % and 90 % of heat from the fuel energy in the cylinder bore is input into circulation. Another important indicator of the combustion process is the variability of individual consecutive cycles. The coefficient of variation (COV_{IMEP}) is defined as the fluctuation of the indicated mean effective pressure (IMEP), which is used as a measure of cycle to cycle variability. It is a common variable for combustion stability in a spark-ignition engine. The combustion of an SI engine is considered to be stable if the COV_{IMEP} is lower than 10 %. The COV_{IMEP} for 200 successive engine cycles at each operating conditions was calculated by Equation 3.

$$COV_{IMEP} = \frac{\sigma_{IMEP}}{IMEP} \cdot 100\% \quad (3)$$

where

COV_{IMEP} ...coefficient of variation of the indicated mean effective pressure[%],

σ_{IMEP} standard deviation of IMEP [bar].

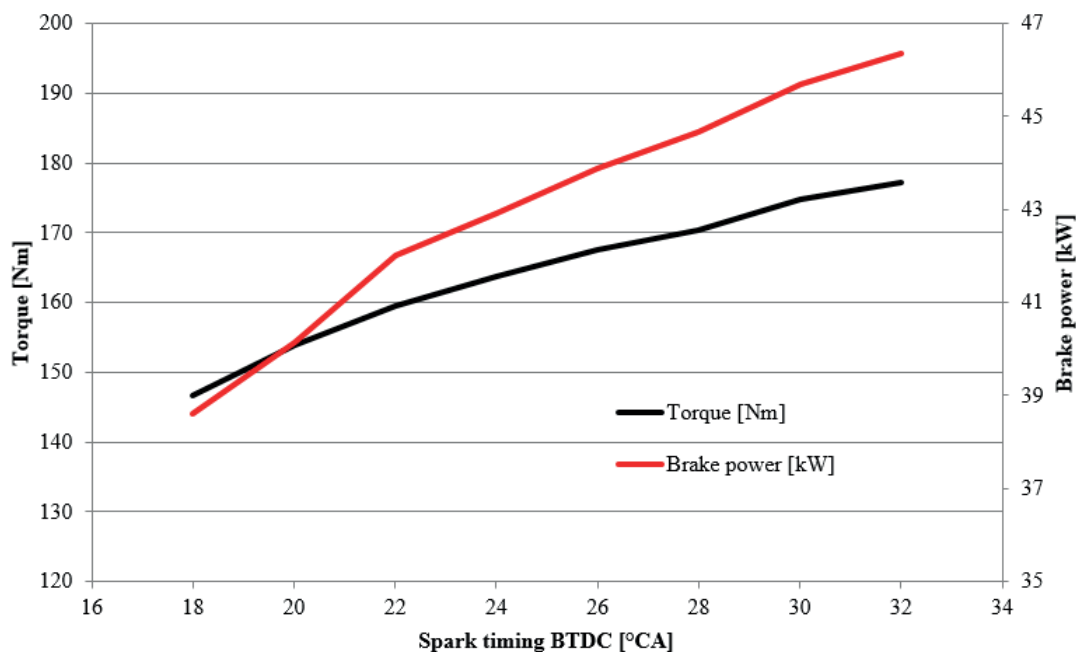
The output engine parameters (power and torque) were recorded during the last 15 seconds of each measurement point. The engine combustion

analysis includes data recorded from 200 cycles of the engine (one cycle = four strokes of the engine). Results represent mean values.

RESULTS AND DISCUSSION

This part of the article contains measured results, according to the above mentioned measurement methodology, which are graphically illustrated in several figures. The results of the output parameter measurements are shown in the load (adjustment) characteristic. The ignition timing is an independent variable, and the engine power and torque are dependent variables. A graphic illustration is shown in Fig. 1.

The graph above shows that the ignition timing affects both measured parameters. As the ignition timing increases, the output parameters of the engine also increase. The highest values are achieved at a maximum ignition timing angle. The results of the adjusting characteristic indicate that the ignition of the mixture in cylinder 32 °CA BTDC results in better conversion of the chemical energy contained in the fuel into mechanical work. Therefore, the best combustion and maximum pressure in the cylinder should be achieved at this ignition timing value. This assumption will be confirmed or refuted by the curves of the cylinder pressure variation. Binjuwair and Alkudsi (2016)



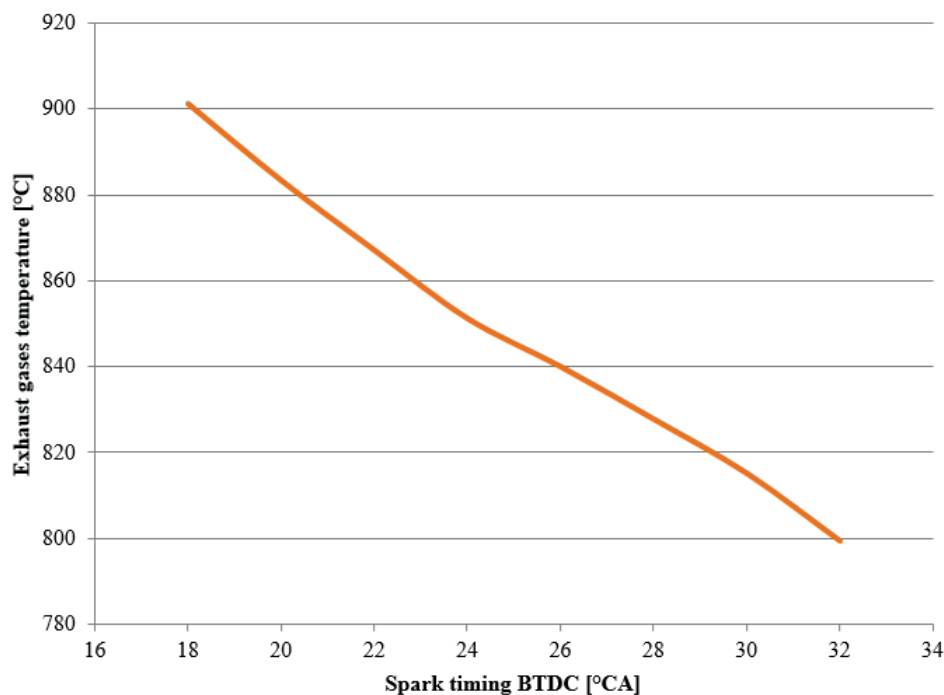
1: Variation of brake power and torque with different spark timings

published similar experimental measurements for a four-stroke single-cylinder engine with the same compression ratio, engine speed and at full throttle. The engine geometry parameters per cylinder (bore, stroke, displacement) and ignition timings are almost identical. The authors came to similar conclusions, namely that earlier ignition of the mixture in the cylinder has a positive effect on the output parameters of the engine. The moment when the spark jumps between the spark plug's electrodes also affects the temperature of the exhaust gases. Higher exhaust temperatures are caused by late ignition and therefore a late onset of combustion of the mixture in the expansion stroke, causing afterburning in the exhaust stroke, all the way in the exhaust pipe. This adversely affects the thermal efficiency of the engine, because the thermal energy released by the burning mixture in the cylinder rapidly escapes through the exhaust pipe, which means it is not optimally converted into mechanical work. Heat losses increase and the thermal efficiency of the engine decreases. (Shi *et al.*, 2016) The output parameters of the engine decline for the same reason (Fig. 1). The influence of ignition timing on exhaust gas temperature can be seen in Fig. 2.

The plot shows that increasing ignition timing negatively correlates with the temperature of exhaust gas. The difference between the highest and lowest ignition timing value is 100°C. The optimal engine operating mode under the given operating conditions is achieved with an ignition timing of 32 °CA BTDC (Fig. 1 and 2). As one of the results of their experimental measurement, Binjuwair

and Alkudsi (2016) also published the effect of the ignition timing on the thermal efficiency of the engine. They found that earlier ignition of the mixture in the cylinder increases the thermal efficiency of the engine, where the minimum and maximum angle of the ignition timing resulted in a 5 % difference.

The effect of the ignition timing on the combustion process in the engine cylinder is demonstrated most significantly by the curves of the cylinder pressure. The cylinder pressure variation is the most important parameter of the thermodynamic analysis of the engine. It is a representative indicator of the entire combustion process, which also allows us to assess the efficiency of energy conversion in the engine. (Bennett *et al.*, 2016) For most applications, combustion analysis data is shown relative to top dead centre (TDC) of the power stroke. The signal level and the variation relative to the position of TDC are important in this regard. The graphical plots are shown in Fig. 1. It is clear that the ignition timing significantly affects the cylinder pressure variation, its maximum values and the location of pressure peaks relative to the TDC of the piston. The late ignition of the mixture in the cylinder (near the top dead centre) on the compression stroke results in a very small increase in combustion pressure in the expansion stroke (see cylinder pressure curves for ST 18 and 20 °CA BTDC). This means that the chemical energy contained in the fuel is not optimally converted into mechanical work. The reason for this is the rapid escape of the combustible mixture through the exhaust pipe, meaning that a part of



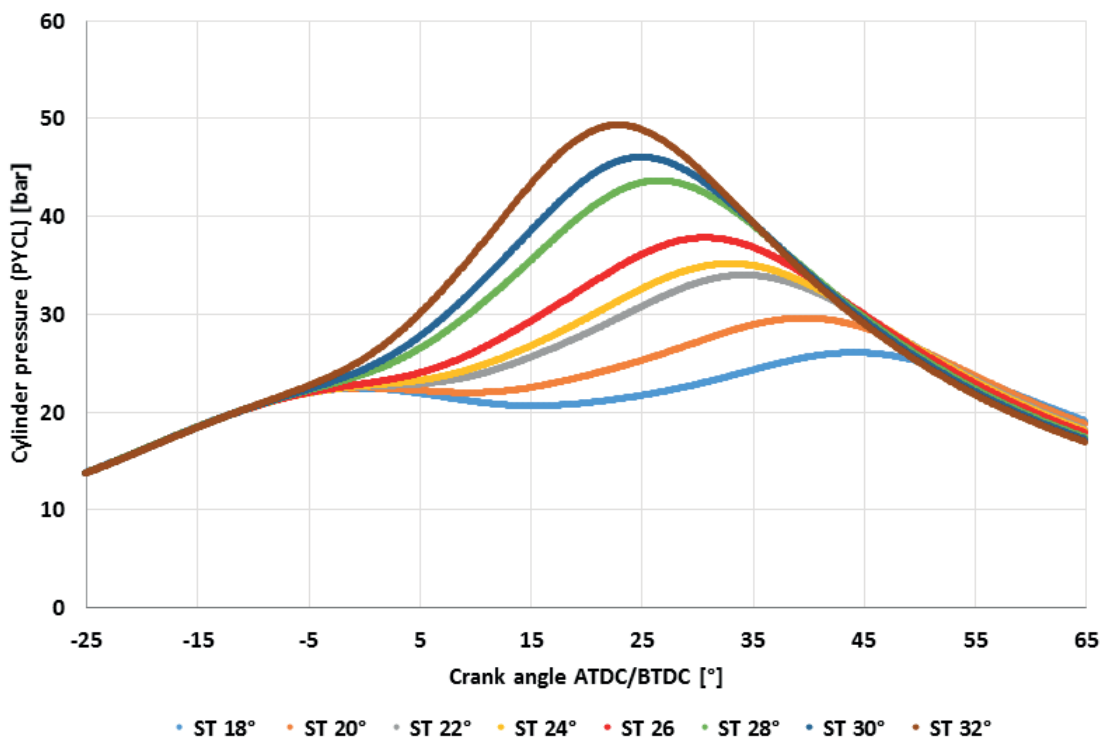
2: The dependency of spark timing on exhaust gas temperature

the combustion will only take place in the exhaust stroke. The related increase in exhaust gas is shown in Fig. 2. This phenomenon adversely affects fuel economy, emissions, the overall engine efficiency and ultimately its service life. The measured results correspond with the cylinder pressure variation published by Binjuwair and Alkudsi (2016). A greater ignition timing angle results in a higher increase in pressure in the cylinder and pressure peaks closer to the top dead centre in the expansion stroke. The thermal energy released by the burning mixture and the subsequently formed pressure acts on the piston throughout the entire expansion stroke, resulting in highly efficient total energy conversion and increased engine efficiency. The engine achieves the best cylinder pressure variation at an ignition timing of 32 °CA BTDC, which confirms the hypothesis established based on the results in Fig. 1.

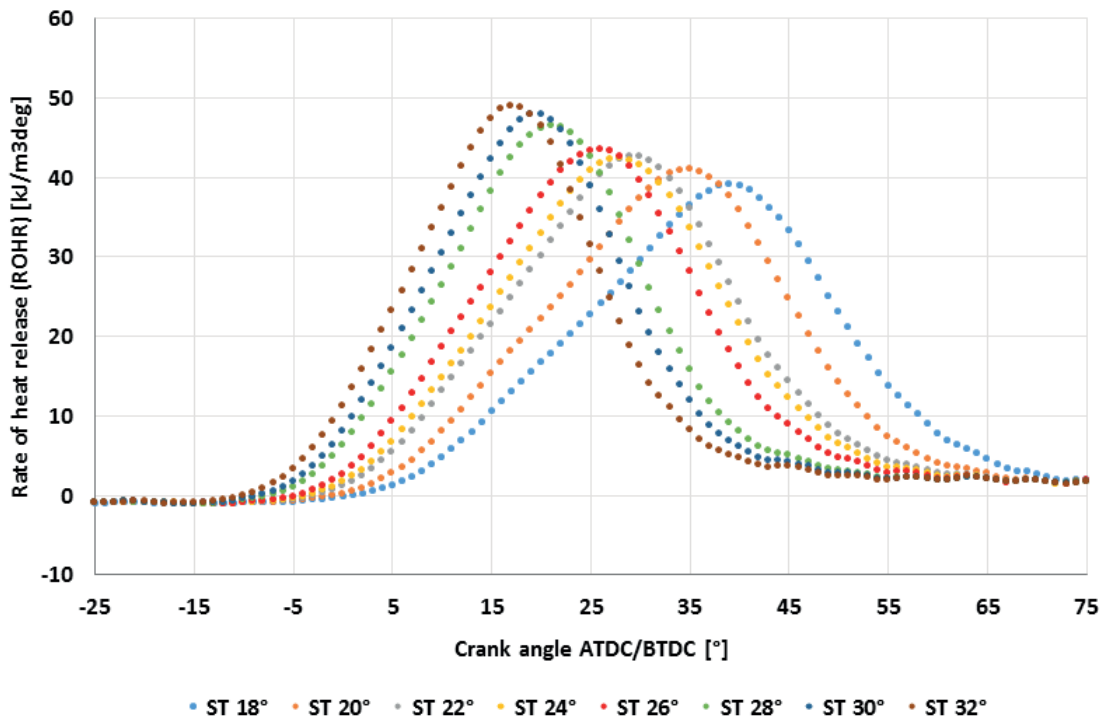
Fig. 4 shows the rate of heat release (ROHR) in the engine cylinder. This is the amount of heat released relative to the cylinder volume, with respect to the angle of the rotation of the engine crankshaft. It is calculated indirectly from the cylinder pressure curve and volume based on the first law of thermodynamics, according to equation (2). The late ignition of the mixture results in lower ROHR values, whereas their peaks are achieved further away from the top dead centre in the expansion stroke in comparison with earlier ignition timing. The ROHR curves also provide information about the conversion of energy in the engine. The curve peaks indicate the position

of the piston relative to the top dead centre (based on the crank angle), where the energy in the fuel is maximally converted to heat and pressure energy. As in the previous graphic plots, the best results are achieved at ST 32 °CA BTDC. By integrating the rate of heat release curves, we obtain cumulative heat or cumulative mass fraction burned (MFB). Its variation determines the beginning (MFB 5%) and end (MFB 90%) of combustion, based on the heat supply into the circulation of the fuel energy in the combustible mixture. Based on the knowledge of these factors, we can also specify the time within which combustion takes place in the engine cylinder. To express the heat supply in circulation, the position of the crankshaft is also determined, in which 10% and 50% of the fuel energy is introduced into circulation in the cylinder relative to the top dead centre (MFB 10% and MFB 50%). The MFB 50% corresponds with the ROHR curve peaks. The graphical waveforms of the integral generated heat can be seen in Fig. 5.

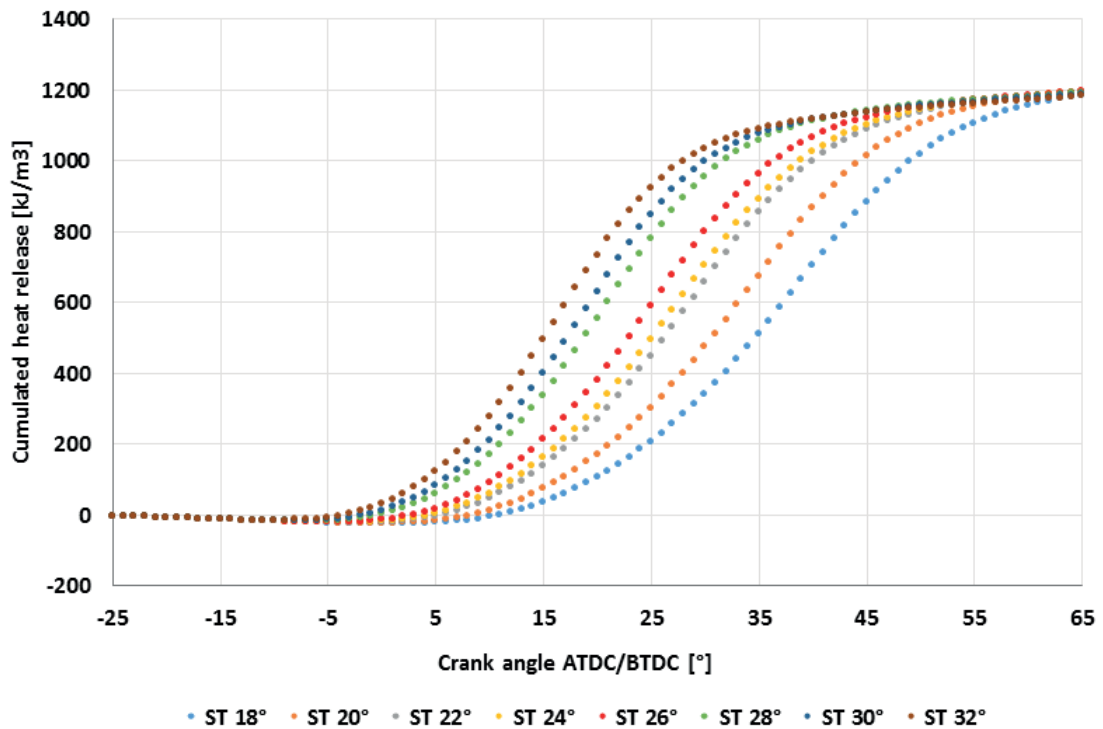
The graph above shows that decreased ignition timing results in later combustion in the engine cylinder. The amount of released thermal energy by the burning mixture is approximately the same for all timings. The energy that is converted to mechanical work of the crank mechanism is important for the efficient operation of the engine, but it cannot be determined in this graphic plot. The ROHR curves in Fig. 4 are relevant for assessing this phenomenon; they show that the decreasing ignition timing angle adversely affects the maximum value of generated heat. The reason for this is



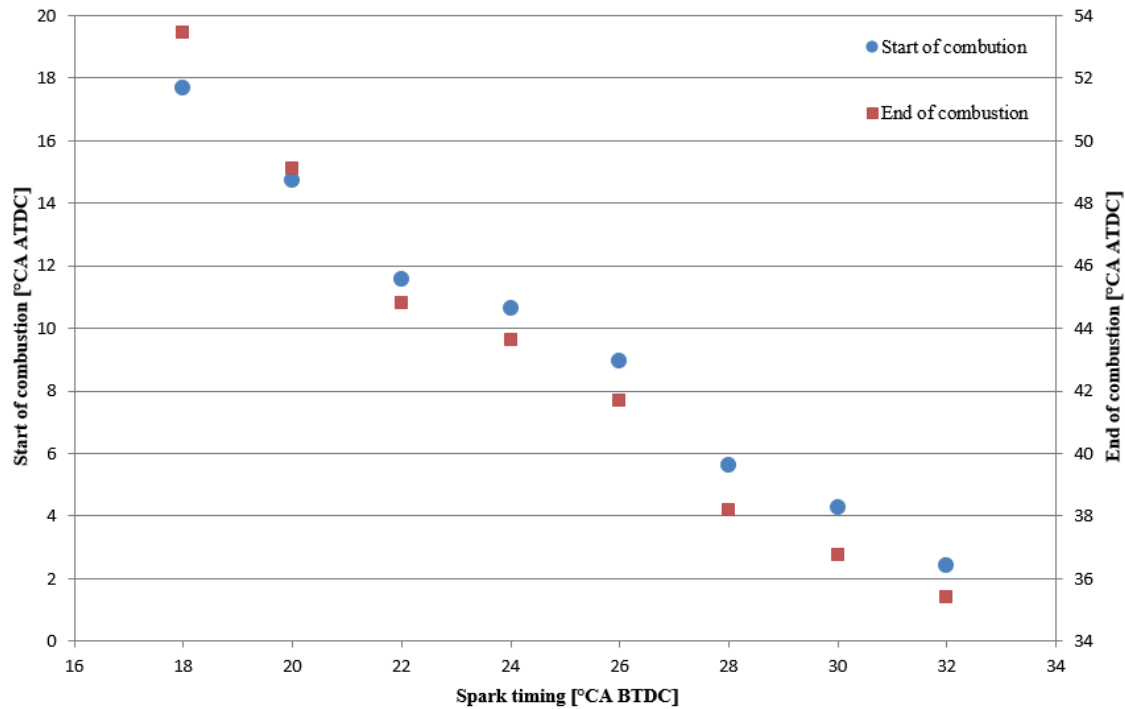
3: Development of cylinder pressure at different spark timing



4: ROHR variations for different spark timings



5: Cumulated heat release for different spark timings



6: Start (MFB 5%) and end (MFB 90%) of combustion for different spark timings

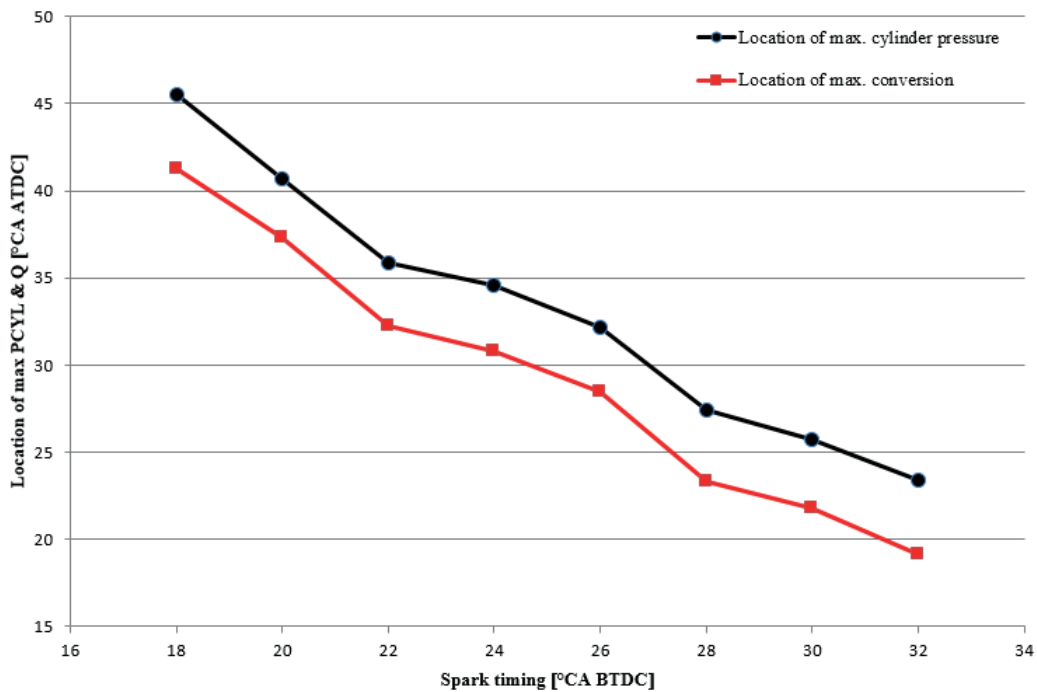
the rapid escape of thermal energy into the exhaust pipe, increasing the engine losses and reducing its thermal efficiency.

Fig. 6 shows the degree of rotation of the engine crankshaft at the upper dead centre of the piston, corresponding with the beginning and end of the combustion of the mixture in the engine cylinder. These parameters are determined from MFB curves, as was mentioned above. The difference between the beginning and end of combustion will give us the duration of the combustion in degrees of rotation of the crankshaft. The graph shows us that the duration of combustion is not very different for each ignition timing, since the maximum difference is only 3 °CA. The total duration of combustion ranges from 36 °CA for ST 18 °CA BTDC to 33 °CA for ST 32 °CA BTDC. In terms of complete fuel combustion, it is desirable for the combustion to begin closest to the top dead centre of the piston in the expansion stroke. 32 °CA BTDC fuel ignition is therefore satisfactory, because combustion already occurs at 2 °CA ATDC.

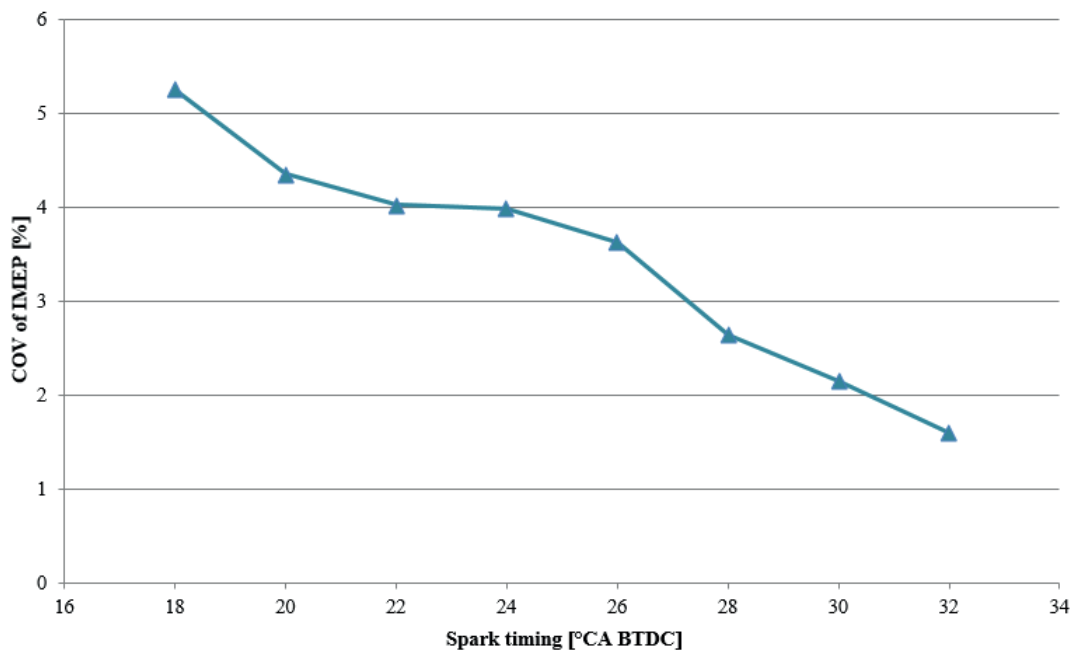
For the optimal conversion of energy in the engine and for increasing its efficiency, the degree of rotation of the crankshaft relative to the top dead centre of the piston is important, where the cylinder pressure (max. PCYL) and generated heat peaks are achieved (max. Q), which corresponds with the maximum conversion of fuel energy to mechanical work. These two parameters are shown in Fig. 7. As the ignition timing angle increases, the PCYL and Q peaks approach the top dead centre of the piston in the expansion stroke, which is desirable for optimum utilization of the chemical

energy in the fuel. The difference between the lowest and highest ignition timing is 20 °CA. If the cylinder mixture is ignited at 32 °CA BTDC, the heat generation peak occurs at 19 °CA ATDC, whereby the cylinder pressure peak is achieved at 23 °CA BTDC.

The final evaluated dependency is the effect of the ignition timing on the stability of the combustion process. The fluctuations of the indicated mean effective pressure (IMEP) is monitored in consecutive engine cycles. The measure of this parameter is therefore the coefficient of variation of the indicated mean effective pressure (COV_{IMEP}). The lower the value, the more stable the combustion. If the value is below 10%, the combustion is considered stable. The COV_{IMEP} curve in relation to the ignition timing is shown in Fig. 8. We can therefore say that the combustion process in the engine cylinder is considered stable for all ignition timings. A slight variation of the IMEP occurs in the consecutive cycles with the increasing angle of the ignition timing. The most stable combustion is achieved at ST 32 °CA BTDC. In their study, Binjuwair and Alkudsi (2016) also published this graphic plot and came to similar conclusions.



7: Location of maximal cylinder pressure and conversion for different spark timings



8: The coefficient of variation of IMEP for different spark timings

CONCLUSION

The aim of the performed measurements was to analyze the combustion process with a change in the ignition timing of the fuel and air mixture in a spark-ignition engine. The measurement results showed that as the ignition timing increases, the engine power and torque also increase. The increase in these parameters is a reflection of higher pressure in the cylinder, the maximum value of which is achieved at higher ignition timing near the top dead centre in an expansion stroke. In these conditions we can expect higher engine efficiency. We also found that the combustion is more stable with

a higher value of ignition timing. No significant difference was found in the duration of combustion at different ignition timings.

The indicated results show that the ignition timing has a significant effect on the output parameters of the engine. Thanks to optimization of combustion, it is possible to achieve higher combustion efficiency and fuel economy. Changing these parameters is also likely to be positively reflected in the volume of produced exhaust emissions. The final question is to what value the ignition timing can be increased. An excessive increase in the ignition timing may result in detonation when using fuels with a low octane rating. Determining the effect of high ignition timing values on the presence of detonation in fuels with different octane ratings will be the subject of further measurements.

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