Abstract. The results of analysis of thermal cycle of the test engine are presented in the paper. The study focused on determining the ignition delay in compression ignition engine. The correlations available in literature, Hardenberg and Hase, Wolfer and Watson and Assanis were used to determine ignition delay. With the increase of the EGR the ignition delay has increased. It turned out that very often it is necessary to determine own ignition delay correlation.

Key words: ignition delay, combustion, modeling, engine

INTRODUCTION

Ignition delay is one of the most important parameters of diesel engines which will directly affect the performance, emissions and combustion. A number of investigations have been conducted to study the ignition delay of diesel fuel. The results showed that the ignition delay depends on fuel parameters and pressure, temperature and excess air fuel ratio.

Rodriguez and all [3, 2, 9] in their work the results of engine tests of biodiesels obtained by transesterification of palm oil and rapeseed oil and with fossil diesel fuel as a reference have presented. Palm oil and rapeseed oil biodiesel gave shorter ignition delay than fossil diesel fuel due to the higher cetane number for the biodiesels. The ignition delay data were correlated as a function of the equivalence ratio, the mean cylinder pressure and mean temperature over the ignition delay interval. A comparison was made with other available correlations [3]. In their study the correlation for predicting the ignition delay of two biodiesels in a direct injection diesel engine was developed. The start of combustion was estimated using the pressure rise curves. At each condition, ignition delay was determined as the difference between start of injection and start of combustion. The new proposed correlations for biodiesels have been compared against the Watson and Assanis correlations. The comparison of results showed that the new correlations predict ignition delay for biofuels better than the available correlations for diesel fuel. It is therefore concluded that the new correlations significantly improve the ignition delay predictiveness for biodiesels in a wide range of parameters such as the cylinder pressures and temperatures at injection, the equivalence ratio, and the engine load and speed conditions [3]. Alkhulaifi and Hamdalla [4] in their work results of studies of ignition delay are presented. Watson, Assanis, Hardenberg and Hase correlations have been developed based on experimental data of diesel engines. However, they showed limited predictive ability of ignition delay when compared to experimental results. The objective of the study was to investigate the dependency of ignition delay time on engine brake power. An experimental investigation of the effect of automotive diesel and water diesel emulsion fuels on ignition delay under steady state conditions of a direct injection diesel engine was conducted [4, 12]. The ignition delay experimental data were compared with predictions of Assanis and Watson ignition delay correlations. The results of the experimental investigation were then used to develop a new ignition delay correlation. The newly developed ignition delay correlation has shown a better agreement with the experimental data than Assanis and Watson when using automotive diesel and water diesel emulsion fuels especially at low to medium engine speeds at both loads.

In addition, the second derivative of cylinder pressure which was the most widely used method in determining the start of combustion was investigated [4, 16, 17]. Zou and all in their work the ignition delay of a dual fuel engine operating with methanol ignited by pilot diesel have investigated. The experimental results showed that the polytrophic index of compression process of the dual fuel engine decreases linearly while the ignition delay
A number of parameters directly affect the ignition delay process during the delay period of the dual-fuel engine. As the engine speed, the ignition delay of the dual-fuel engine by time scale (ms) decreased clearly under all engine operating conditions. However, the ignition delay of the dual-fuel engine increased remarkably by advancing the delivery timing of pilot diesel, especially at light engine loads [11, 24, 26]. Liu and Karim [13] changes in the physical and chemical processes during the ignition delay period of a gas-fueled diesel engine (dual-fuel engine) due to the increased admission of the gaseous fuels and diluents has examined. The extension to the chemical aspects of the ignition delay with the added gaseous fuels and the diluents into the cylinder charge was evaluated using detailed reaction kinetics for the oxidation of dual-fuel mixtures at an adiabatic constant volume process while employing n-heptane as a representative of the main components of the diesel fuel. The extension to the chemical process of the ignition delay, which results from the chemical interactions between the diesel and gaseous fuels, was the main rate-controlling process during the delay period of the dual-fuel engine. The extent of the extension to the ignition delay period depends strongly on the type of the gaseous fuel used and its concentration in the cylinder charge [13]. In spark ignition engines, under defined conditions, a self-ignition of the air-fuel mixture can occur, too [15, 18]. This phenomenon is known as knock combustion [19, 22].

Many researches around the world are involved in the modeling of the process of combustion in compression ignition engines [23, 28, 14, 18]. It is advanced computer programs that are used for this purpose, which serve for solving flows in combustion engine chambers of any geometry by numerical methods. These are programs belonging to the fluid mechanics field, where numerical methods are employed for solving CFD (Computational Fluids Dynamics) problems. One of them is AVL Fire.

THEORETICAL APPROACH

The ignition delay is the time between the start of injection and the start of combustion. It is widely accepted that the ignition delay has a physical and a chemical delay. The physical delay is the time required for fuel atomization, vaporization and mixing with the air, whereas the chemical delay is the pre-combustion reaction of fuel with air [1]. Ignition delay in diesel engines has a direct effect on engine efficiency, noise and exhaust emissions. A number of parameters directly affect the ignition delay period, among them cylinder pressure and temperature, swirl ratio and misfire. In addition to these effects the recent trend of changing fuel quality and types has a great effect on ignition delay. Experimentally, the start of ignition is mainly determined by the first appearance of visible flame on a high speed video recording [5], or sudden rise in cylinder pressure or temperature caused by the combustion [6, 25, 27].

In the literature there are many correlations for predicting ignition delay. They exist as a function of engine and charge parameters. This correlation can be written of the form:

$$\tau_{id} = A p^{-n} \exp \left( \frac{E_A}{RT} \right)$$  \hspace{1cm} (1)

where: $E_A$ – apparent activation energy for the fuel auto ignition process, $R$ – universal gas constant, $A$ and $n$ – constants dependent on the fuel.

The ignition delay is defined as the time between the start of fuel injection and the start of detectable heat release [1]. The ignition delay is a function of mixture pressure, temperature, excess air ratio. In an engine, pressure and temperature change during the delay period due to the compression resulting from piston motion. To account for the effect of changing conditions on the delay the following empirical integral relation is usually used [1]:

$$\int_{t_{si}}^{t_{si}+\tau} \left( \frac{1}{\tau} \right) dt = 1,$$  \hspace{1cm} (2)

where: $t_{si}$ – the time of start of injection, $\tau_{id}$ – the ignition delay period, $\tau$ - the ignition delay at the conditions pertaining at time $t$.

Well-known correlation describing the ignition delay is formula developed by Hardenberg and Hase for the duration of the ignition delay period in DI engines [2]:

$$\tau_{id} = (0.36 + 0.22c_{in}) \exp \left[ E_A \left( \frac{1}{RT} - \frac{1}{17190} \right) \left( \frac{21.2}{P-12.4} \right)^{0.67} \right]$$  \hspace{1cm} (3)

where: $\tau_{id}$ – ignition delay in crank angle degrees (CA), $T$ – temperature in K, $P$ – pressure in bars, $E_A$ – the apparent energy activation, $c_{in}$ – the mean piston speed (m/s), $R$ – the universal gas constant (8.3143 J/molK).

The activation energy is given by:

$$E_A = \frac{618840}{CN + 25}$$  \hspace{1cm} (4)

where: $CN$ – the fuel cetane number;

Another correlation proposed Wolfer [9], Watson [10]:

$$\tau_{id} = 3.45 \frac{\exp \left( \frac{21000}{T} \right)}{P^{1.82}}$$  \hspace{1cm} (5)

where: $P$ – pressure, $T$ – temperature.

The correlation proposed by Assanis [7] is a function of equivalence ratio, with a pre-exponential factor of 2.4
that considers the equivalence ratio variations (from 2.6 to 3.8 for the combined pre-exponential factor) as opposed to the Watson correlation where it is fixed at 3.45. This constant value used by Watson would translate into 

\[ \tau_{id} = 2.4 \left( \frac{2100}{T} \right)^{0.07} \Phi^{0.2} \].

In AVL Fire the ignition delay is calculated on the basis of correlation [8]:

\[ \tau_{id} = 4.804 \cdot 10^{-8} \left( \frac{N_{O_2}^{n,M}}{n,M} \right)^{0.53} \left( \frac{N_{Fe}^{n,M}}{n,M} \right)^{0.07} (\rho)_{i,3}^{0.13} e^{\frac{-5914}{T}} \]

where: mole fraction of species (O\(_2\), fuel) is in mol/m\(^3\), temperature in K, density \(\rho\) in kg/m\(^3\).

### Table 1. Modeling parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine rotational speeds</td>
<td>1500 rpm</td>
</tr>
<tr>
<td>Cylinder bore</td>
<td>107.19 mm</td>
</tr>
<tr>
<td>Crank throw</td>
<td>60.325 mm</td>
</tr>
<tr>
<td>Connecting-rod length</td>
<td>245 mm</td>
</tr>
<tr>
<td>Initial pressure for 180° CRA before TDC</td>
<td>0.16 MPa</td>
</tr>
<tr>
<td>Initial temperature for 180° CRA before TDC</td>
<td>310 K</td>
</tr>
<tr>
<td>Injection angle</td>
<td>-9° CA</td>
</tr>
<tr>
<td>Injected fuel mass</td>
<td>0.0735 g/cycle</td>
</tr>
<tr>
<td>Injection duration angle</td>
<td>20° CA</td>
</tr>
<tr>
<td>Fuel temperature</td>
<td>330 K</td>
</tr>
<tr>
<td>FIRE program's sub-models</td>
<td>k-zeta-f</td>
</tr>
<tr>
<td>Turbulence model</td>
<td>Coherent Flame</td>
</tr>
<tr>
<td>Combustion model</td>
<td>ECFM-3Z Model</td>
</tr>
</tbody>
</table>

### RESULTS AND DISCUSSION

The paper presents results of 3D modeling of engine thermal cycle operating at a constant rotational speed.
The work investigates the influence of EGR on engine operating parameters, on heat release rate and ignition delay. The study was conducted for constant injection timing and load. Research engine is supercharged engine. After the validation process of the model were started modeling. Highly compatible modeling and experimentally obtained results (Fig. 2) was received. Calculated engine efficiency is the gross efficiency, the modeling does not include charge exchange loop.

Figure 3 shows the influence of EGR on the mean indicated pressure and indicated efficiency of the test engine. At the time of increase the participation of the EGR mass fraction of fuel injected into the cylinder was constant. At 60% share of the EGR the biggest drop in efficiency was received, and it reached a value of 36%. Similarly, the value of indicated pressure was decrease to value equal 1.16 MPa. There has not been optimization of the thermal cycle of the test engine.

Figure 4 shows a comparison of the rate of heat release curves. One line represents heat release rate of modeled engine and the others traces were obtained on the basis of experimental studies. The curves of heat release rate were used to determine the ignition delay. The algorithm for determining the ignition delay was presented in the literature review.

Figure 5 shows the lines for ignition delay calculated using Hardenberg and Hase, Watson and Assanis correlations and determined on the basis of modeling results. In conditions without EGR, the closest value of ignition delay was obtained by using of the Assanis correlation. With the increasing of the recirculated exhaust gas participation, the ignition delay increased, which is consistent with results obtained by other authors. The results obtained with the use of Hardenberg and Hase and Assanis correlation did not give satisfactory results.
With the increase of the EGR to 60% the ignition delay value has increased from 3.1 to 5.2 deg. Figure 6 shows the cross sections, in two planes, of the combustion chamber. The reaction rates of combustion in the modeled combustion chamber are presented. It is clear that in the case with EGR the space covered by combustion process in the combustion chamber is smaller than in the case without recirculation. This is due, inter alia, by the ignition delay.

CONCLUSIONS

The results of analysis of thermal cycle of the test engine are presented in the paper. The study focused on determining the ignition delay in compression ignition engine. Ignition delay is one of the most important parameters of diesel engines which will directly affect the performance and emissions. In order to determine this parameter, the correlations available in literature were used. The ignition delay on the basis of modeling results was determined, too. It turned out that, in this case, these correlations did not give satisfactory results. With the increase of the EGR to 60%, the ignition delay value increased from 3.1 to 5.2 deg, in the model test engine.

Based on the literature study it can be said that other authors also stated that these universal correlations usually do not give the expected results. Often it is necessary to determine the correlation by defining ignition delay.

REFERENCES


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