INTERACTION BETWEEN INJECTION TIMING, EGR – RATE AND RATE OF HEAT RELEASE IN THE TDI – ENGINE

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Abstract
The high-speed diesel engine has been developed intensively in the past few years. Customer-relevant engine performance data such as BMEP, fuel consumption, exhaust and noise emissions have been improved and the advantages in comparison to the gasoline engine (fuel consumption and torque characteristics) has become more significant. To comply with the very low soot and NOx – limits diesel engines operate with high injection pressure, high EGR – rate and very late injection timing. As these low emissions are the result of different Rates of Heat Release (ROHR), it has to be investigated which features of ROHR are relevant to emissions. They provide important information as to the direction that further development work should take.
The investigations were made for broad parameter variation at different engine operating points. As one result it can be pointed out that ROHR itself is not sufficient to describe emissions. At some operating points with different A/F ratios the ROHR were the same and emissions were different.
However, ROHR in connection with A/F – ratio and other thermodynamic parameters give very good information on NOx and soot emissions which indicates the practical possibilities to reduce these emissions.

1. Introduction
During the past 10 years TDI engines in passenger cars have made steep progress in customer-related performance values: power density increase from 35 to 58.2 kW/l, maximum BMEP up to 21.2 bar, combustion noise reduction by application of pilot injection, exhaust emissions from EG1 to D4 levels, increase in efficiency up to 42.8 % with additional measures on the power train and car body (one-liter car). Variable turbine geometry and high-pressure fuel injection (unit injector up to 2000 bar) are the basic new technologies underlying this progress.
Taking into account that the exhaust emissions have been reduced by more than 90% during the past 30 years [1] it becomes obvious that the biggest steps have already been taken, for both, gasoline and diesel engines. Future scenarios demand further efforts. In order to stay competitive the development of specific power and BMEP, especially at low revolutions, is necessary. However, there exists a trade-off between specific power output and exhaust emissions (Fig. 1-1).
Measurements which reduce just one of the components of exhaust emissions are not useful, such as for example particulate trap systems which also cause an increase in fuel consumption. For this reason, research work on the in-cylinder combustion process with the target of simultaneously reducing exhaust emissions and fuel consumption is essential.
2. Combustion process of different diesel engines

The combustion process of modern diesel engines starts later than it did in the past. The reason for this are the legislative NO$_x$ - limits which can be achieved by lowering the flame temperature. One method is the retardation of injection timing.

The cylinder pressures of some direct-injection diesel engines (different manufacturers) have been compared to demonstrate the combustion evolution in the past years.

The engines are all water-cooled, high-speed, direct-injection diesel engines (Table 2-1); three of them are 4-cylinder engines with a cylinder-volume of about 400-650 ccm, two others are 1-cylinder research engines with a cylinder-volume of 511 and 1850 ccm, respectively. All engines were measured at approx. 2 bar BMEP at 1900-2000 rpm or at 1200-1600 rpm, where the 1200-value corresponds with 50% of speed range in SBP and SBG-engine.

**Table 2-1.**

<table>
<thead>
<tr>
<th>Type of engine</th>
<th>DI diesel engines, water-cooled</th>
</tr>
</thead>
<tbody>
<tr>
<td>Symbol</td>
<td>PD</td>
</tr>
<tr>
<td>No. of cylinders</td>
<td>[-]</td>
</tr>
<tr>
<td>Cyl. volume ca.</td>
<td>[cm$^3$]</td>
</tr>
<tr>
<td>Compression</td>
<td>[-]</td>
</tr>
<tr>
<td>Torque at engine speed</td>
<td>[Nm] [rpm]</td>
</tr>
<tr>
<td>BMEP</td>
<td>[bar]</td>
</tr>
<tr>
<td>Inlet closed</td>
<td>[°C]</td>
</tr>
<tr>
<td>Outlet open</td>
<td>[°C]</td>
</tr>
<tr>
<td>Fueling</td>
<td>Unit</td>
</tr>
<tr>
<td>Fuel consumption</td>
<td>[kg/h]</td>
</tr>
<tr>
<td>Air consumption</td>
<td>[kg/h]</td>
</tr>
<tr>
<td>Inlet pressure abs.</td>
<td>[bar]</td>
</tr>
</tbody>
</table>
The results of cylinder pressure measurements are presented in fig. 2-1. It is easy to observe that in research 1-cylinder engines (AV03, SBP, SBG) the pressure rise occurs much earlier (nearly isochoric) than in 4-cylinder engines where combustion starts at 7-18 degrees after TDC. Measurements marked as AV03 were carried out on the modern-design engine with 4-valve cylinder head and medium supercharging, but with the injection time optimized for better overall efficiency, not for emission.

![Fig. 2-1 Indicated pressure of different diesel engines with different injection timing](image)

In contrast to this, the standard-production or near standard-production engines (TD, CT and PD) must operate with later combustion (nearly isobaric) to comply with the NOx – limits. It should be pointed out that the newer the engine is, the later the combustion will run. Unfortunately, this is connected to a great rise in fuel consumption, CO and HC emissions. On the other hand, the injection timing cannot be shifted beyond a limited BMEP standard deviation. For this reason, new methods of emission reduction must be developed. But at first the mechanisms which lead to the emissions must be understood. This paper is a small step in this direction and deals with the interactions between engine parameters (Start of Injection, Exhaust Gas Recirculation) and soot and NOx emissions. The investigations were made by a modern TDI engine at four operating points, relevant in the NEDC.

3. Interaction between ROHR, soot and NOx emissions with injection timing variation

Fig. 3-1 shows the ROHR values at 2000 rpm, BMEP=2 bar without EGR and at different SOI. In the lower half of the diagram Specific Fuel Consumption and emissions can be seen. The classical two phases of diesel combustion appear only until SOI= –6 deg. After this crank angle they tend to resemble a homogenous process similar to the spark plug engine. It can be seen that the maximum of ROHR stays constant from SOI=–10 deg. This is caused by two facts: Firstly, if the SOI is later, the maximum rate will be lower as the cylinder volume increases very fast. That makes the combustion slower. Secondly, more energy has to be delivered to enrich the same load, as the thermal efficiency at late combustion is very poor and
the maximum of ROHR rises again. As a combination of these two effects the maximum of ROHR stays constant.

![Figure 3-1](image1.png)

**Fig. 3-1** ROHR, SFC, soot and NO\textsubscript{x} emissions with variation of SOI at 2000 rpm and BMEP= 2 bar without EGR

However, it is much more interesting that from SOI=−10 deg the soot and NO\textsubscript{x} emissions decrease simultaneously. Unfortunately, the fuel consumption and the emissions of HC and CO\textsubscript{x} which are not shown here, rise. These emissions can be easily reduced with a catalytic converter but the higher fuel consumption cannot be improved as the thermal efficiency at late SOI is very poor. As this behavior is typical for most operating points in the NEDC it should be investigated in detail. It is inherent to the combustion at late injection timing that both the soot and NO\textsubscript{x} emissions are reduced simultaneously.

While the reduction of NO\textsubscript{x} emission at late injection timing is well known, the soot emission should normally be higher, as seen in the classical soot – NO\textsubscript{x} trade-off.

The reason for decrease of soot at late injection timing is the ignition delay. It is shown in **Fig. 3-2** together with the soot emissions for a variation of the SOI. If the SOI is delayed, injection and combustion become more and more separated. In the extreme case the whole fuel is delivered before start of combustion. The result of this is avoiding fuel injection in the flame and that is most important to reduce the soot emission [2]. It can be seen that maximum soot emission appears for the same SOI as the minimal ignition delay. **Fig. 3-2** also shows that there are other parameters which have an influence on soot emission. The ignition delay at SOI=−14 and −6 is the same, but the soot emission value is nearly two times as high.
It is a known fact that the local parameters are the most important parameters for emissions, for example the maximum flame temperature or the local A/F ratio [3]. But these parameters are very difficult to estimate. This is why we attempt to investigate with this work whether there are global thermodynamic parameters which in addition to further measured values provide the possibility to describe soot and NOx emission. This would deliver information as to how the ideal combustion should run to reach the best possible compromise between emissions and fuel consumption.

4. Correlation between thermodynamic parameters, soot and NOx emissions

It is another known fact that ROHR has an influence on emissions. However, the global parameters are not responsible for emissions as soot and NOx depend on local parameters such a local A/F ratio or flame temperature which are different to the global A/F ratio or global temperature. For this reason it is regarded as impossible to determine a general equation describing the emissions from ROHR. But for a similar kind of combustion (same engine, same injection system), the local parameters relevant for emission are connected with the global parameters. The model in Fig. 4-1 shows this. There is an elementary reaction zone in an area with global thermodynamic parameters for two cases. The combustion runs in the reaction zone which surrounds a fuel droplet (not drawn). The fuel from this droplet evaporates and thus the available chemical energy for the reaction is delivered to the reaction zone.
On the left hand side of the figure the case without EGR is shown. The partial oxygen pressure is high and the specific heat capacity is low. This increases the flame temperature because the surrounded gas cannot take a high amount heat and the combustion runs very fast. The high flame temperature and high oxygen partial pressure cause a strong NO – formation. The conditions shown on the right hand side of the figure are in contrast to this. There is a high amount of inert gas, such as CO₂ and H₂O, and a small quantity of oxygen which cools the reaction zone very well. Apart from this, the combustion process takes more time. As a result the flame temperature is lower. In connection with a smaller quantity of oxidizer a poor NO formation takes place.

This way, emissions of the elementary reaction zone depend – apart from other factors - on global parameters. However, with the same parameters the summarized emissions can be different if there are more elementary zones in the combustion chamber or if they release more heat. This occurs when the fuel droplets inside the reaction zone are bigger. Therefore, the ROHR itself is not sufficient as the same ROHR at bigger cylinder volume causes other emissions than a smaller volume. So the other thermodynamic parameters have to be taken into account to describe combustion and emissions unequivocally. But the question is: Which parameters should be considered? First of all, the parameters cannot be connected with the crankshaft (center of combustion...), as this does not provide enough information: “to lower the soot and NOₓ emissions the injection timing has to be later” will be the only result.

Having analyzed the possible scenarios, six global parameters were chosen (fig 4-2). A/F ratio and ml/mg inform about mass of oxygen and inert gas outside the reaction zone. dQ_max specifies how much heat is released in the reaction zone and V_dQB_max describes the dilution of gas outside the reaction zone. The T_SOC and

Where:

- T_SOC: Temperature at start of combustion
- V_SOC: Volume at start of combustion
- dQ_Bmax: Maximum of ROHR
- V_dQBmax: Volume at maximum of ROHR
- ml/mg: Ratio of air to total gas mass

**Fig. 4-2 Parameters describing the combustion**

**Fig. 4-3 Soot emission in dependency of maximum ROHR (for different operating points)**
$V_{\text{SOC}}$ values describe the thermodynamic parameters at start of combustion. The attentive reader will have noticed that the ignition delay which was shown in a previous chapter as being very important for the soot emission, does not appear. The reason is that there are complications in measuring the injector nozzle lift of a unit injector. But the injection delay depends on the thermodynamic parameters. The most important parameter is the temperature at SOC - $T_{\text{SOC}}$. If this temperature is lower, the ignition delay is longer and the portion of fuel which is injected in the flame is lower. The influence of the defined parameters on the emissions cannot be determined separately as each variation of engine parameters simultaneously causes a change of many defined parameters. For example, if the start of injection is varied, the temperature at start of combustion, and maximum of ROHR etc. will change, too. For this reason the effect of individual parameters cannot be determined. Fig. 4.3 shows this for the independence of soot emission on maximum of ROHR. For the same maximum of ROHR the soot can change about 10 times.

In order to solve this problem, the following procedure has been created. We assumed that soot and NO$_x$ emissions depend on defined parameters, but the intensity and direction of each parameter is unknown and can be different. This can be expressed with exponents which can have both positive and negative values (fig. 4-4).

If this formula includes all parameters which are important for soot and NO$_x$ emission it might be possible to describe the emission for many operating points with the same constants. However, first the constants must be found. To achieve this the data of approx. 140 measured points were considered. There were 4 relevant operating points in the NEDC test, at 3 EGR rates and at approx. 10 different SOI.

The constants were varied until the error sum of squares reached the minimum. The values of constants for NO$_x$ and soot are also shown in fig. 4-4.

The agreement between measured and calculated emissions is shown in fig. 4-5.
First of all, the very good correlation with the calculation of NO\textsubscript{x} emission can be seen. This proves that the major parameters for the NO\textsubscript{x} emission are apparent in the formula. In comparison to this, the accuracy for the calculation of the soot emission is not satisfactory. Two possible reasons for that behavior can be pointed out at the moment. Firstly, not all important parameters for soot emission have been incorporated into the formula. Secondly, the measurement of low soot emission is very inaccurate. Repeated measurement of the same operating point also yields great deviations. For this reason, it could be said that the won formula found for soot emission is rather good.

The subject for future work will be studies on the sensitivity of soot and NO\textsubscript{x} emissions to the change of the presented parameters. It will indicate the possibilities to decrease simultaneously soot and NO\textsubscript{x} emission.

5. Summary

It has been shown that the low emissions of modern diesel engines are connected with other types of heat release processes as it was in the past. In this work we have investigated whether, and if so, which global parameters are essential in calculating soot and NO\textsubscript{x} emissions. Such parameters were found and it was possible to calculate the emissions without using local parameters which are very difficult to determine. The correlation for NO\textsubscript{x} calculation is about 0.99 and for soot 0.81. The poor value for soot could be connected with the inaccuracy of the measurement. In the next work the formulas derived for NO\textsubscript{x} and soot will be used to investigate how combustion should be run to reduce emissions.

References


Fig. 4-5  Agreement between measured and calculated emissions for 140 measured points