

## **A PRACTICAL APPROACH IN THE THERMODYNAMICAL ANALYSIS OF THE TDI - PROCESS**

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**Abstract.** High speed diesel engines have been developed intensively in the past few years. The customers relevant engine performance data as: bmep, fuel consumption, exhaust and noise emissions have been improved. To reach all objectives in time, all affords of designers and scientists with the aid of a modern development will be necessary.

This paper deals with new methods and theories in the thermodynamical analysis of the TDI engine, which need a cylinder indicating pressure, optical measurements and new calculation models.

To evaluate the rate of heat-release, which is introduced into the 2-zone-model, the measured cylinder pressure has to be adjusted. With the aid of a new procedure the absolute level of the measured pressure can be determined. In addition to that an improved method for evaluation of the cylinder-mass in the EGR-mode will be presented.

The combination of the calculated rate of heat – release and measured values from the test cell like the composition of exhaust gas will be read into a new theory of two zones.

The concentration of NO and soot will be calculated in dependence of the crank angle and verified by optical measurements.

The goal of this research work is to reduce the exhaust gas emissions through a better understanding of the internal combustion process. With the so obtained results it would be able to modify engine control and design.

The report about the use of the new methods and theories during thermodynamic analysis at the TDI-research will be continued.

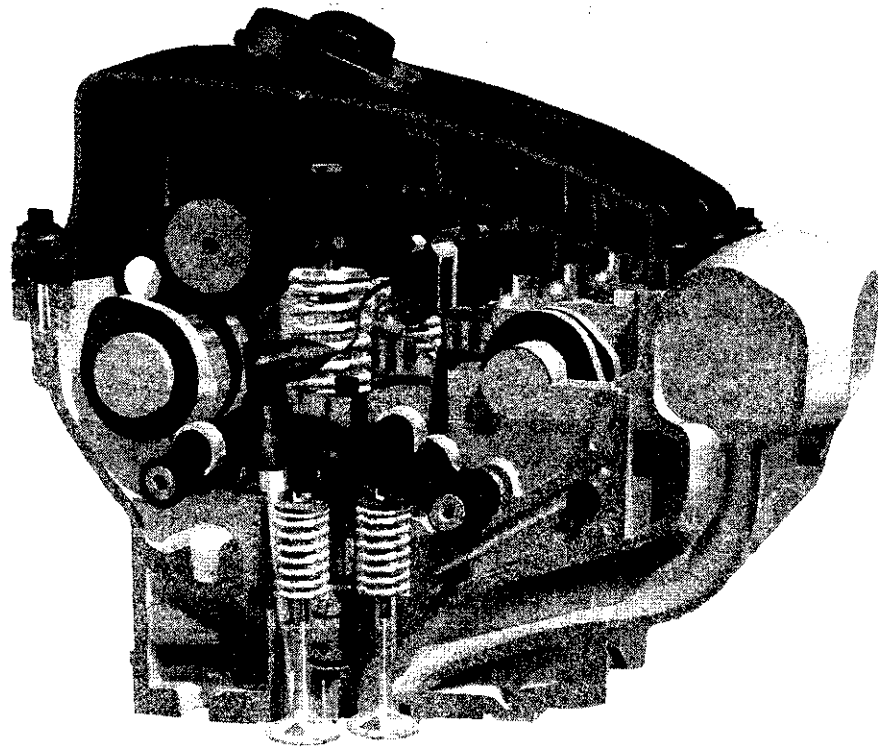
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## **1. Introduction**

In the year 2001 Volkswagen Diesel engines for passenger cars and vans have a tradition of 25 years with increasing success. The actual power range has a spread from 45 kW-3-cylinder to 230 kW-V10-cylinder. Some milestones are set with these engines, for example:

- The first 3 l car (1.2 l-3 cylinder engine)
- The first EU 4 exhaust emission Diesel car (1.2 l-3 cylinder engine)
- Maximum specific power output 58,2 kW/l (1.9 l-4 cylinder engine)
- Maximum b.m.e.p. 20,2 bar (1.9 l-4 cylinder engine)
- The most powerful passenger car Diesel engine, 230 kW / 750 Nm (5 l-V10 cylinder engine)



**Fig. 1: Cylinder head configuration of the advanced Volkswagen TDI engine**

With the exception of cold start the engine noise is nearly equal to gasoline engines. The main problem of today's Diesel engines is the compliance with EU 4 emission standards with the complete engine and car family without losing power output or increasing fuel consumption.

The solution will be found in two ways.

1. Optimising the combustion process to avoid the formation of pollutants at the source
2. Exhaust aftertreatment in order to clean the exhaust gas by further reactions.

The evolution of the combustion process involves the main parameters of the Diesel engine:

- Gas flow in the combustion chamber
- Fuel injection
- Geometrical combustion chamber configuration

One example of this efforts shows **fig. 1**.

For big size engines in heavy cars Volkswagen is developing a combined exhaust aftertreatment system consisting of catalysts and a particulate trap.

The objective of this paper is to lower the NO- and soot emissions during the combustion process. The new tools for this affords are:

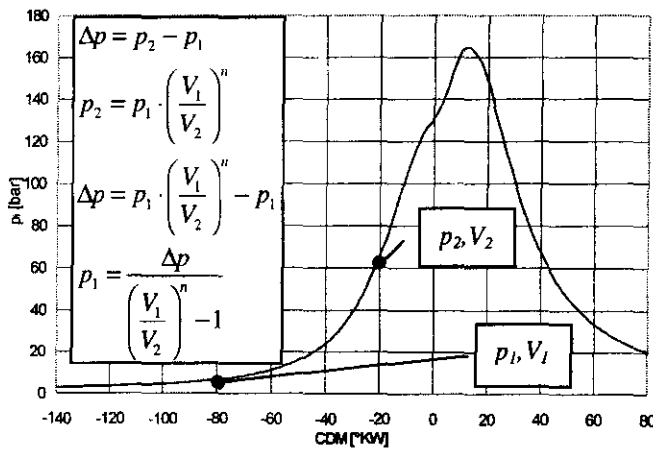
1. Integrated optical measurements in the hardware at the test bench
2. Integrated calculation software in the improved thermodynamic analysis.
3. The strength of these tools is the interactive connection of calculation and measurement.

## 2. Preparation of the measured cylinder pressure to calculate the heat transfer

The cylinder pressure is measured with piezo – transducers. With these sensors it is possible only to measure the relative pressure. For the determination of the absolute pressure level further rake procedures are necessary.

### 2.1. Thermodynamic determination of the absolute pressure level

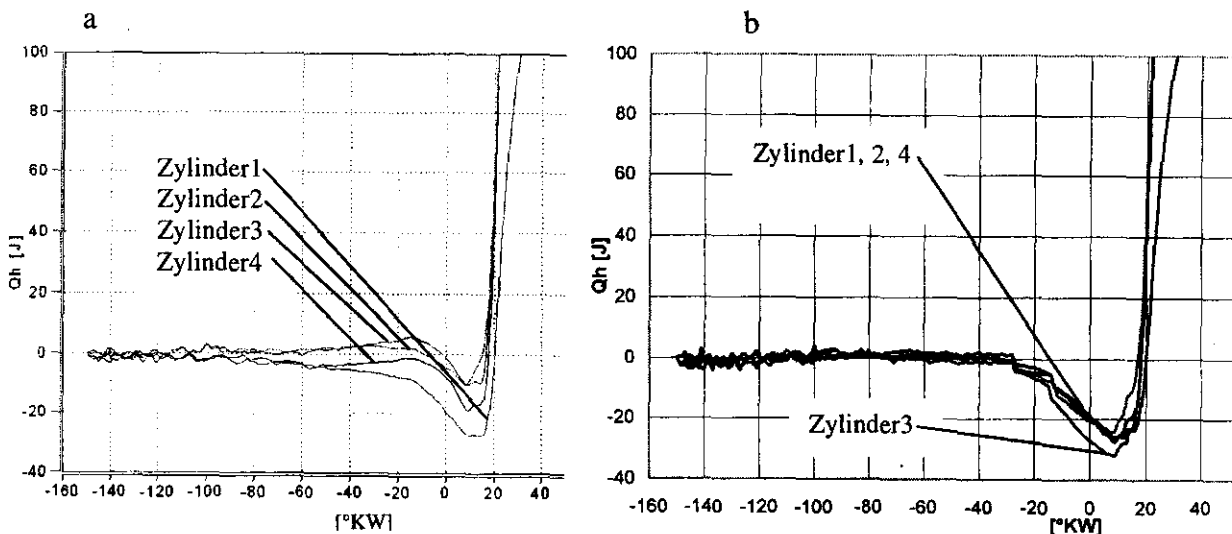
These methods use the features of the polytropic equation. If the polytropic exponent between two crank angles is known the absolute pressure level can be calculated from the difference of the pressures according to the **fig.2**. This procedure is used very often and indicating systems can apply it automatically. But the precise estimation of the absolute pressure level demands an exact knowledge of the polytropic exponent, which depends on the operating point of the engine. For this reason this method isn't sufficient for a precise calculation of the heat transfer.



**Fig. 2: Thermodynamic determination of the absolute pressure level**

### 2.2. Adjusting of the pressure on the intake-manifold pressure

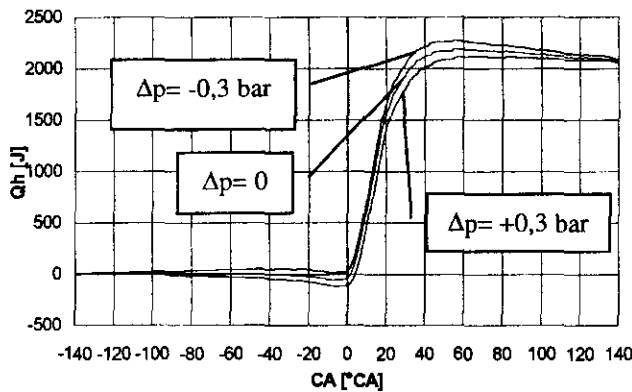
With this method the measured pressure at  $-180^\circ$  CA is compared to the pressure in the intake-manifold. If the pressures are different, the indicated pressure is shifted. The dynamic processes in the intake pipe falsify the measurement of the pressure and produce the differences between the intake pipe pressure and cylinder pressure. This procedure therefore, is only applicable in a limited scale.



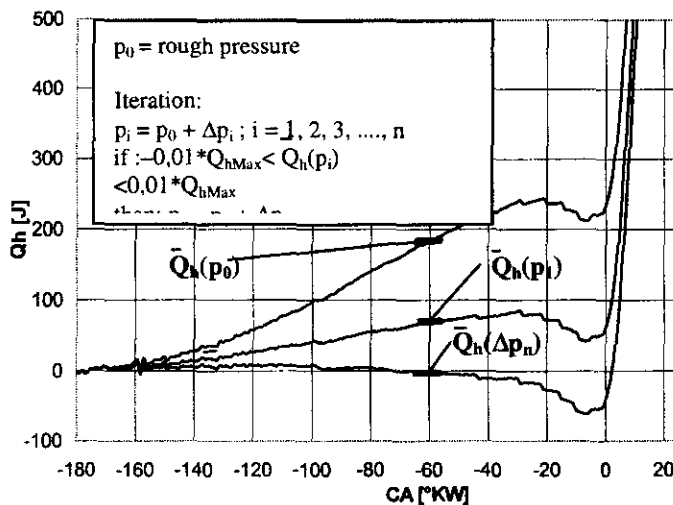
**Fig. 3: Heat function during compression: a) adjusted to the intake manifold pressure and the same compression ratios for all cylinders, b) with individual compression ratios and the new method of determination of the absolute pressure**

### 2.3. Determination of the absolute pressure level by the analysis of heat function

By the calculations of the heat release according to the shown procedures it often appears that the graph increases or declines during compression very intensive. This couldn't be explained by warming up of the gas by the cylinder walls or wall heat losses, because both processes are very poor compared to the total energy fraction. **Fig. 3** shows the described situation. At the operating point of  $2000 \text{ min}^{-1}$ , 10% throttle, the heat function of all cylinders were determined with a commercial software (**a**). In this case the absolute pressure was adjusted on the intake manifold pressure according to chapter 2.2. It is to be seen that the curves are very different during the compression. The reasons for this are at first the errors by the determination of the absolute pressure, because the pressure in every cylinder is not equal with the intake manifold pressure, which is a mean value. In addition to that the commercial software works with a constant compression ratio for all cylinders. But in reality the compression ratios of different cylinders are generally different. The heat functions from the same operating point are shown in **fig. 3b**, but with individual



**Fig. 4: Heat function from 3 pressure levels at  $1900 \text{ min}^{-1}$ , full throttle**

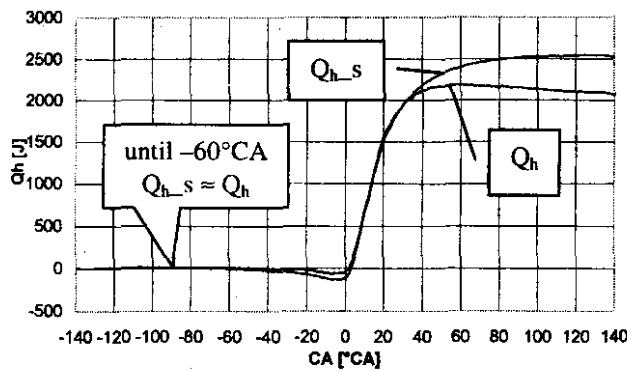


**Fig. 5: Determination of the absolute pressure level according to the heat between  $-55$  and  $-65^\circ \text{CA}$**

compression ratios for each cylinder and a new method of the determination of the absolute pressure. As expected the graphs for all cylinders are the same until  $-30^\circ \text{CA}$ , the warming up from the wall can't be seen neither. Only after  $-30^\circ \text{CA}$  the differences are bigger. The third cylinder has a by far larger compression ratio. Therefore probably higher wall heat losses, too. By the determination of the absolute pressure level the sensitiveness of the heat function against the pressure level

can be used. When the pressure is to low or to high the heat function will go up or down, according to **fig. 4**. So it is possible to determine the pressure level according to the heat function. But it has to be investigated, whether it exists a crank angle near TDC, where the heat is nearly zero. The pressure level has to be changed until the heat function is equal zero in this point. Previous investigations showed that the heat of all operating points is very weak until  $-30 \text{ CA}$ . The warming up from the wall is often not higher than the errors by the pressure indicating.

The higher heat transport to the wall is also seen behind  $-30^\circ \text{CA}$ .



**Fig. 6: Heat function and substitute heat function for the determination of the absolute pressure level**

variable specific heat capacity has to be evaluated. For this reason, a different way was searched. It was found that the modified Hohenberg's equation corresponds to the heat function until  $-60^{\circ}\text{CA}$  very well.

$\Delta Q_{1,2}$  heat energy between 1 and 2

$c_v$ ..... specific heat capacity at constant volume

$R$ ..... gas constant

$\kappa$ ..... isentropic exponent

$p_1, V_1, p_2, V_2$ ..... pressures and volumes

At all operating points the heat between  $-65$  and  $-55^{\circ}\text{CA}$  was not higher than 1% of the maximum heat release. And this fact can already be employed for the determination of the absolute pressure level. In **fig. 5** the method is explained. In the first step the heat function is evaluated from the uncorrected pressure. The pressure level is then varied until the desired agreement is achieved.

This method is combined with high expenditure, because it demands an iterative calculation of the heat function.

The gas mass has to be known and the Hohenberg's equation reads [1]:

$$\Delta Q_{1,2} = \frac{c_v}{R} \cdot V_2 \left[ p_2 - p_1 \cdot \left( \frac{V_1}{V_2} \right)^\kappa \right] \quad (1) \quad \text{where:}$$

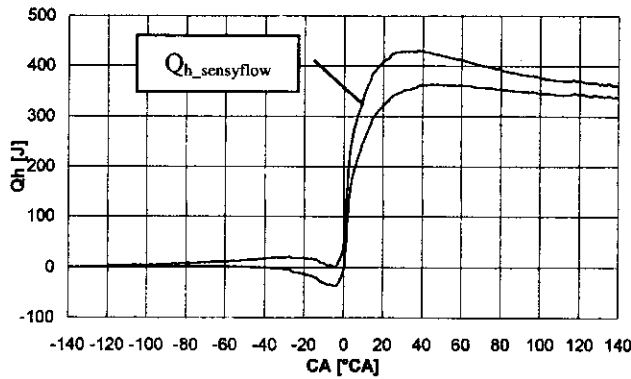
By the calculations of the heat function according to the equation (1) with  $c_v = 0,5 \text{ J}/(\text{kg} \cdot \text{K})$  the best agreement to the real heat release was achieved. This value has no physical means and is used just for mathematical calculations as substitute to the heat function until  $-60^{\circ}\text{CA}$  and can be used for determination of the absolute pressure level (**fig. 6**). The knowledge of the gas mass isn't necessary and the previous calculation can be made with constant specific heat capacity. The course of the curve later than  $-60^{\circ}\text{CA}$  is insignificant; therefore the heat function will be precisely evaluated one more time after the determination of the absolute pressure level.

### 3. Determination of the cylinder – charge with EGR

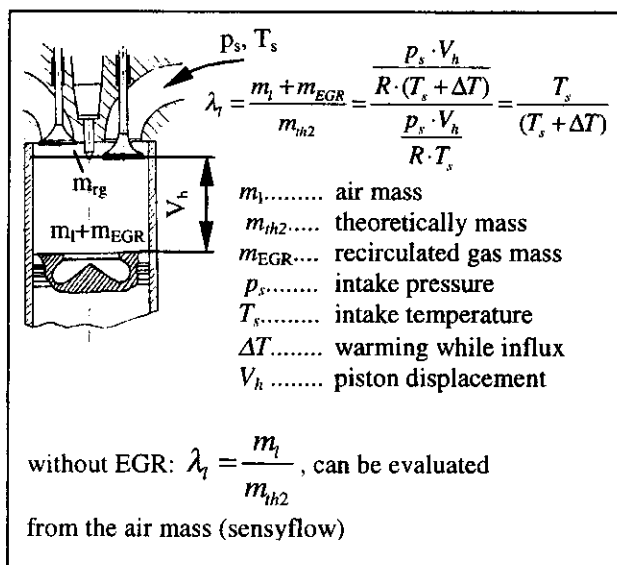
At the investigated engine the cylinder – charge can be evaluated as an air mass plus residual gas, because there is no valves overlap. At the operating points with EGR this method isn't correct, because the exhaust gas is lead back into cylinder. In that case the heat function from this method can't be right of course. It is shown in **fig. 7** in comparison to a different method, which considered the total gas mass in the cylinder and will be described next. When the charge efficiency is known, the cylinder charge can be evaluated from the theoretical mass during filling of the cylinder.

The cylinder charge without EGR results from the next equation:  $m_g = \lambda_1 \cdot m_{th} + m_{rg}$  (2) with:  $m_g$ ... total cylinder mass,  $m_{rg}$ ... residual gas mass,  $\lambda_1$ ... charge efficiency

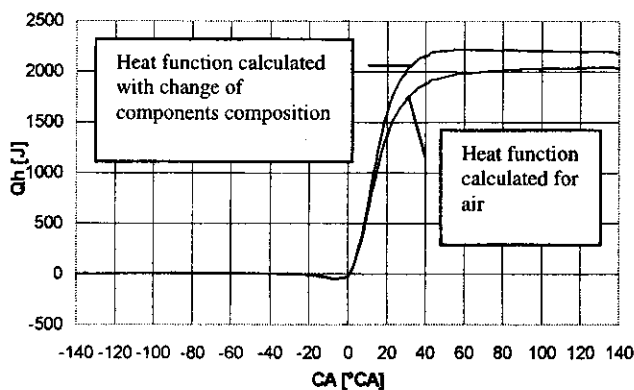
The theoretical mass can always be determined without problems. The only difficulty is the charge efficiency, which depends on the intake flow temperature, according to **fig. 8**.



**Fig. 7: Heat function with EGR calculated with an air mass plus residual gas versus total gas mass including EGR**



**Fig. 8: Determination of charge efficiency**



**Fig. 9: Heat function calculated for air and with change of components composition for an operating point at  $\lambda=1,2$  and  $1900 \text{ min}^{-1}$**

are according to the burning function from the first step (fig. 10). It is iterated with new parameters of the burning function. After three iterations a stable value was reached. In this case the evaluation of the burning function has only three steps.

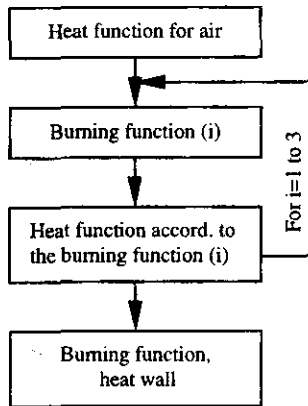
So the charge efficiency from the measurement without EGR can't be used when exhaust gas is recirculated, because the intake temperatures are different. In this case the charge efficiency was evaluated from the measurement without EGR, but at a temperature as high as with EGR. The very high temperature without EGR could be achieved with a water air radiator. The air in this radiator could be both chilled and warmed. Therefore the total gas mass at different EGR – rates can be evaluated from the equation (2) by the knowledge of charge efficiency at different intake temperatures.

#### 4. Calculations of the heat transfer

By the calculation of the heat transfer, except for determination of the absolute pressure level and cylinder charge according to chapters 2 and 3, accuracy is very important by the evaluation of specific heat capacity. The change of components composition can't be neglected, especially at low A/F – ratio, because combustion products have higher specific heat capacity than air.

In the fig. 9 the heat function is shown for calculation with and without change of components composition. It can be seen that the calculation of heat function for air is very inaccurate. So it can't be accepted.

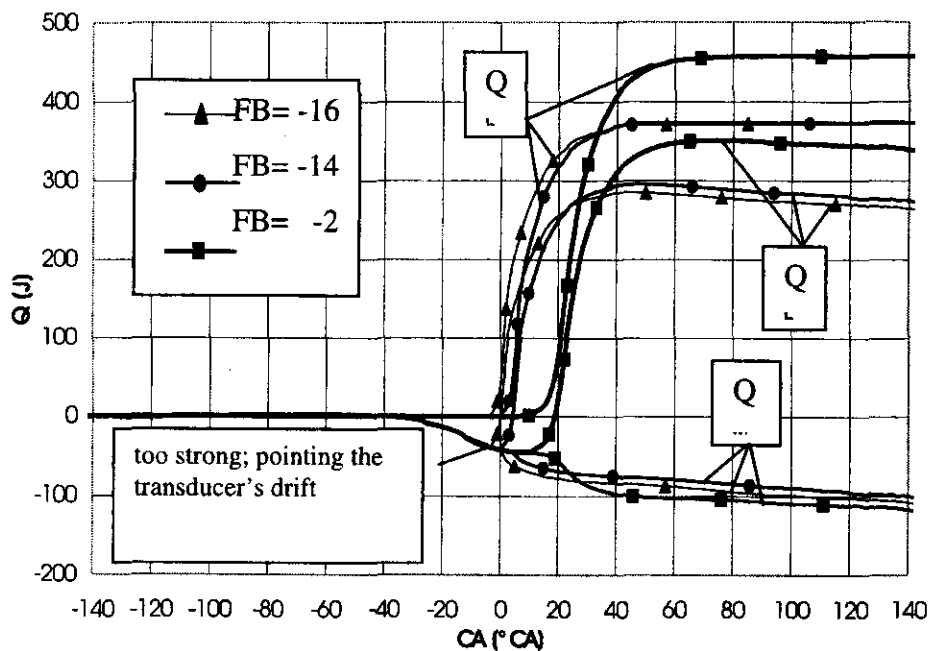
The other curve was calculated iteratively. In the first step the heat function is calculated for air. Then the burning function according to modified Oppenheim's method is evaluated. The Oppenheim correlation [2] is evaluated from minimum to maximum of the heat function. The next calculation of the heat function is made with products, which



**Fig. 10: Evaluation of the heat transfer**

The wall heat losses are determined next as difference between burning function and heat function. In this case the evaluated wall heat losses includes blow – by heat and the evaporation heat of the fuel. These heat energies are very weak, so it can be said, that the difference between burning function and heat function is dominated by the wall heat losses.

**Fig. 11** shows heat transfer at an operating point  $2000 \text{ min}^{-1}$  at  $\text{BMEP} = 2 \text{ bar}$ ,  $\text{EGR} = 15\%$  and three starts of delivery . It is obvious that the calculation of the heat transfer started at the beginning of the compression. The heat function stays until  $-30^\circ\text{CA}$  at the zero line. The warming up from the wall is very weak in comparison with total fuel energy. Further the heat function and wall heat losses increase very strong. If the wall heat losses and the fuel evaporation are taken into account, the difference to the zero line is  $20 \text{ J}$ . So the problems with the small uncooled pressure transducers can be seen. The cause for it is probably drift of transducers, which haven't been researched yet in a high – tech diesel engine. An other research project will investigate these problems.



**Fig. 11: Heat transfer at  $2000 \text{ min}^{-1}$  and  $\text{BMEP}=2 \text{ bar}$ ,  $\text{EGR} = 15\%$  for different injection timing**

The burning function will be next read in 2 – zone – model and the motor results will be analysed.

## 5. Simulation of the exhaust fraction with a 2 – zone model

In the following part some aspects of the formation of  $\text{NO}_x$  and soot in a traditional diesel engine, working at partial load, are discussed. The results of this study were obtained by simulation with a 2-zone model and were confirmed by measurements.

The 2-zone model divides the combustion process in two zones: an unburned zone, which contains the reactants, and a burned zone, which contains the combustion products. An

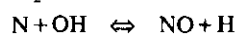
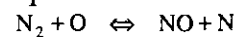
infinite thin flame layer separates the zones. The heat of the combustion is introduced in the second zone and is obtained from the burning rate as calculated in the previous part. From the preservation of energy in the burned zone and the global system the temperature of the unburned zone can be calculated. The model doesn't take any geometric properties of the combustion zone into account. The chemical reactions take place in the burned zone.



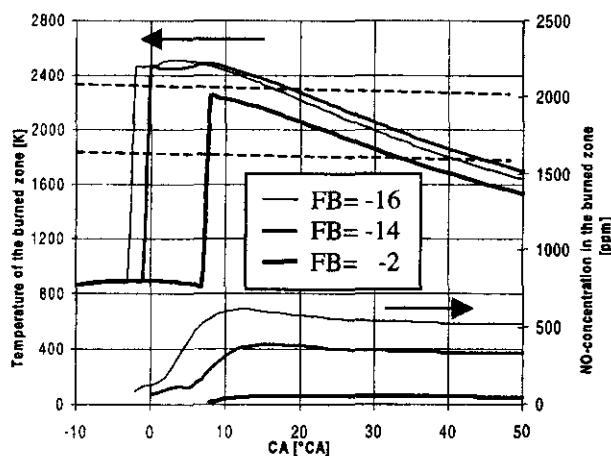
Some of the reactions are considered to take place in chemical equilibrium:

All chemical reactions slow down with decreasing temperatures.

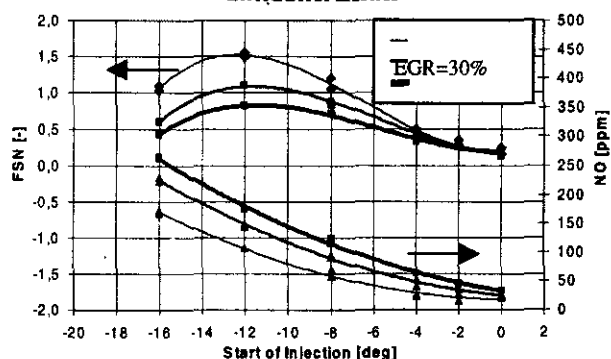
The lower reaction rate is different for each reaction. The NO-formation reaches the chemical equilibrium in a few milliseconds at temperatures higher than 2000K. As the temperature in the internal combustion process of diesel engines is only short time above 2000K, the NO-formation doesn't have time to reach the chemical equilibrium. That is why the NO-formation and decomposition have to be calculated in non-chemical equilibrium. The 2-zone model is equipped with a Zeldovich mechanism. [3]



In **fig. 12** the effect injection timing variation on the temperature of the burned zone and the NO-formation can be seen. At early injection points the temperature of the burned zone is high. When the temperature of the burned zone is higher than 2400K a lot of NO is formed, caused by the very high oxygen concentration is this "temperature window".



**Fig. 12: Maximal temperature and NO - concentration**

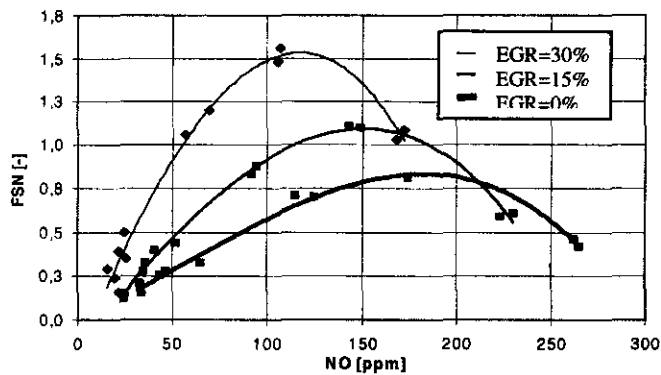


**Fig. 13: Emissions at 2000 min<sup>-1</sup>, BMEP=2 bar**

temperature remains relatively short in the second "temperature window" between 2400 and 1800K, where there is a possibility to decompose NO. When the temperature of the burned zone is lower than 1800K the NO reactions are "frozen". The thermochemistry of the equilibrium products ends at a much lower temperature. From **fig. 12** it can easily be seen, that at later injection timing, the temperature of the burned zone stays relatively short in the NO-formation zone and extremely long in the NO-decomposition zone, leading to very low NO concentrations.

The 2-zone model also includes a new soot model based on the soot model of Hiroyasu. The soot concentration in the cylinder is a function of soot formation and soot decomposition. The combustion process is divided into two phases: one where the burning rate differs from zero and one where the burning rate equals zero. In the first

phase the soot formation is mainly determined by the mass of burned fuel, obtained from the burning rate. In the experiments and calculations presented in [4] it has been shown, that the rate of soot oxidation during the combustion is mainly determined by the OH concentration. In the second phase it is assumed that no new soot is formed. The soot oxidation is mainly a function of the swirl. The empirical parameters in the first phase are fitted to the soot concentration obtained from the multispectral analysis with a multi-colour method. In the second phase, the empirical parameters of the model, when the exhaust valve opens, are adjusted to the soot concentration in the exhaust gas. The calculation results of the 2-zone model are in accordance with the experiments. [3]. From **fig. 13** and **fig. 14** it can be seen that for very late injection timing soot- and NO<sub>x</sub> are simultaneously reduced. This effect is present with and without EGR and is due to partial homogenisation. These phenomenon is the subject of further investigation.



**Fig. 14: NO<sub>x</sub> and soot trade-off**

## 6. Conclusions

- The indicated cylinder pressure trace has to be corrected in order to get a plausible heat function. A new method of adjusting has been presented.
- The problem of evaluating the gas mass in the cylinder when using EGR has been overcome by a new method using the temperature dependency of the charge efficiency
- This improved methods lead to a more accurate burning function which is used as an input for further calculations regarding the thermodynamical analysis.
- The main tool used in this investigations is an improved 2-zone model which is adjusted on results from the test bench gained by an optical measurement, that delivers the radiation energy of the burning gas.
- The main result of this work is the simultaneous reduction of NO<sub>x</sub>- and soot emissions by late injection timing and various EGR rates.

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