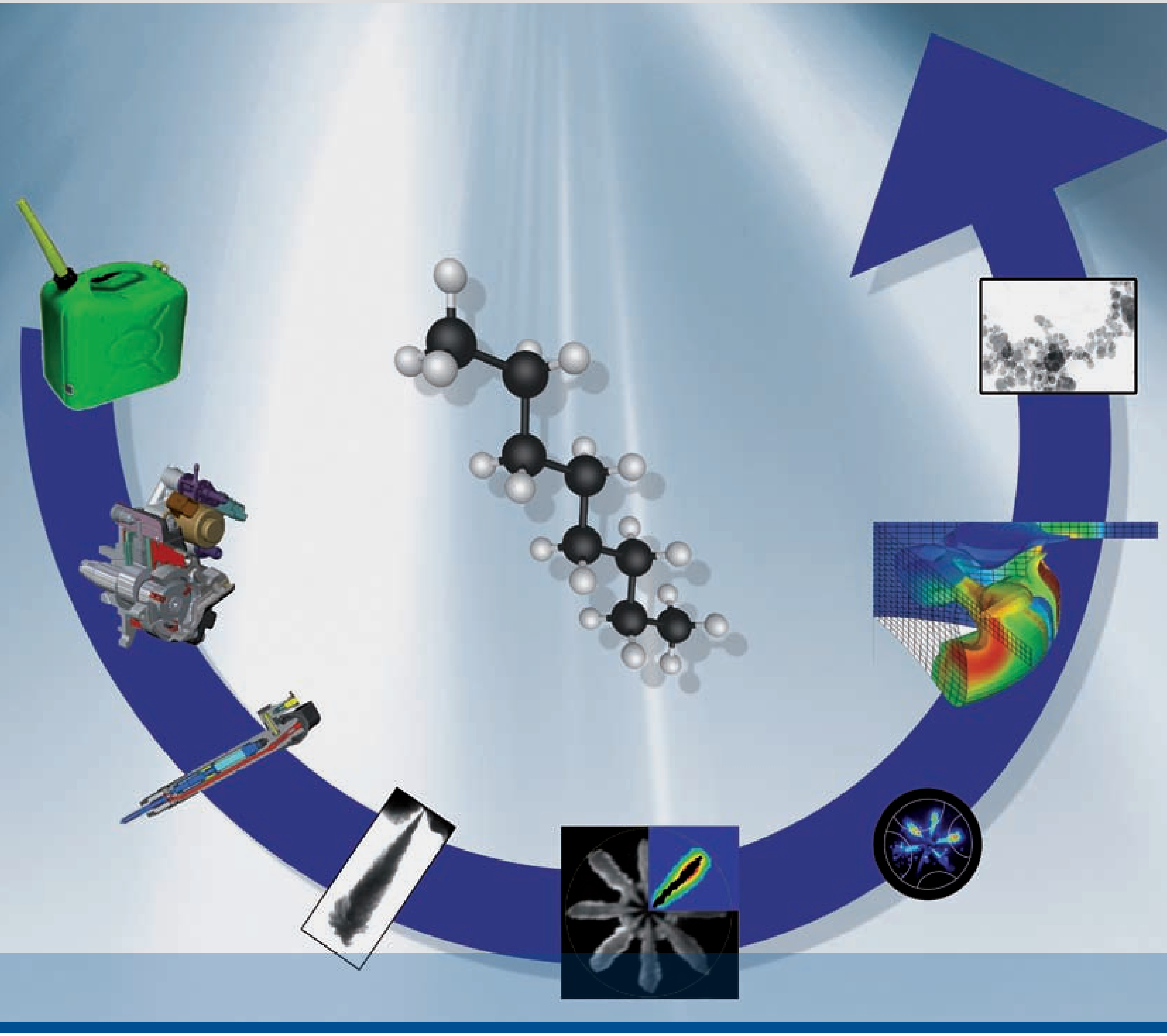

World Wide Fuel Qualities

New Challenges for Combustion Control in Advanced Diesel Engines

Against the backdrop of general discussions on the reduction of CO_2 emissions, diesel engines will face new challenges in the future in order to meet future emission limits worldwide while maintaining good consumption figures, good driving characteristics, and acceptable costs. One of these challenges is that the fuel characteristics of diesel often varies greatly worldwide, and sometimes even within one country. As part of this article, we will illustrate FEV's approach for the compensation of different fuel grades during combustion. The basic influencing factors for a possible compensation of the fuel's influences will be demonstrated and analyzed, and first results for a suitable control concept will be presented.

1 Introduction

By taking a look at the combustion process in diesel engines, it is obvious, that the integration of fuel properties will be of considerably greater importance in the future. This has various reasons. For one thing, there is the desire to replace at least part of the fuel that is based on fossil crude oil with a regeneratively produced portion. This is due to the problem of global warming and the demand for a massive reduction in CO_2 emissions. It is not yet fully clear, which strategy for substitution should and can be pursued. At first, a partial substitution of diesel fuel seems to make the most sense, because the proportion of diesel fuel that is consumed by vehicles on the road is already significantly higher today than that of gasoline fuel, at least in Europe. It seems that this will not change in the future either [1]. One consequence this has is that diesel and gasoline fuel is moved within regions with a different proportion of gasoline and diesel engines in order to keep the efficiency of the manufacturing process in the refinery at a high level. A much higher proportion of diesel vehicles in other markets would limit this practice significantly and, when looking at the overall system, lead to a great loss in efficiency and thus to higher CO_2 emissions during the refining process.

However, a further increase of the portion of diesel substitutes of the so-called first generation bio fuels (e.g. rape

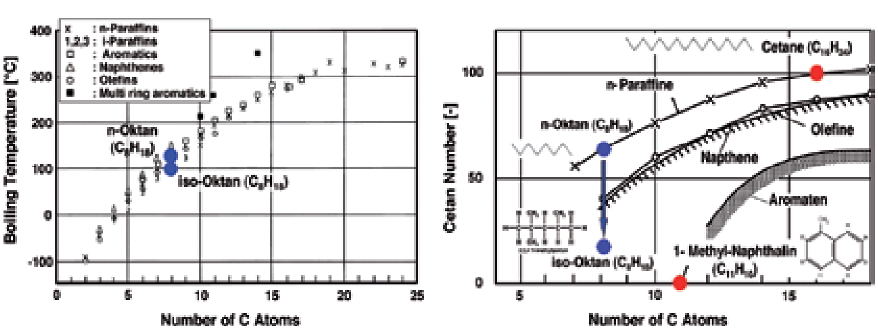

Figure 1: Some fuel properties depending on molecular structure [3]

Authors

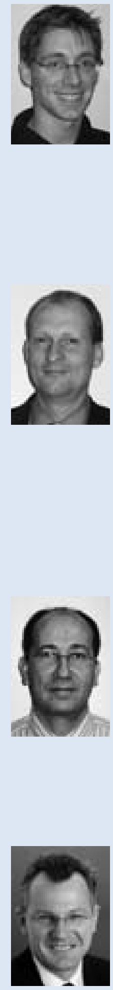

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Table 1: Influence of molecular size and structure on fuel characteristics

	Molecule size	Molecular structure
Vaporization	Increasing boiling temperature with a higher number of C atoms	Negligible influence with regard to the molecular structure
Density	Increasing density with a higher number of C atoms	Cyclical structures have a higher density
Calorific value	Calorific value depends on H/C ratio liquid hydrocarbon fuels (>C4): Hu = 42…44 MJ/kg	
Auto ignition	Increasing auto ignition tendency with a higher number of C atoms	Compact molecules and double bond ➔ reduced auto ignition tendency
Rate of combustion	The number of C atoms and the molecular structure only have a minor influence on the laminar rate of combustion of liquid fuels in air (➔ 40 cm/s)	
Soot formation tendency	Soot formation increases with a higher C/H ratio	Increases with compactness of the structure Paraffins ➔ Naphtenes ➔ Aromatics
Viscosity	Increases with a higher number of C atoms	Increases as follows: Naphtenes➔ Olefines ➔ Paraffins

seed methyl ester, vegetable oil) will not lead to the desired goal, as the savings potential with regard to production and CO_2 is limited [2]. Diesel substitutes of the 2nd generation (Fischer Tropsch fuels) demonstrate a much higher potential here and also have less problems when it comes to long term stability. However, these fuels will in mid term not be available in large quantities.

But all in all it can be stated, that there will be a continuous substitution of fossil diesel fuels. Depending on the fuel characteristics, this will also have an impact on combustion as well as on exhaust aftertreatment.

Another aspect already mentioned earlier is the probable or the intended expansion of the market share of vehicles with diesel engines in other markets such as China, India, USA and Russia. The fuel characteristics sometimes differ greatly between these countries. In the US, a market where vehicles with the most advanced diesel engines will be increasingly offered in the next few years, the fuel grades and properties fluctuate substantially even within the individual states.

These considerations show that future combustion systems must be very robust in spite of further reduced emission limits. Additional advanced control algorithms for the regulation of combustion must at least partly be able to compensate the influence of the fuel within the scope of the particular specified fuel standardization.

As part of this article, a look at basic, currently known influences of different fuel grades on the combustion process in diesel engines will be taken. Two fuels with clearly different specifications as an example to show possible compensation measures through intelligent, self-learning control functionalities will be used.

2 Fundamentals of Diesel Engine Fuels

Petroleum-based fuels are blended fuels and consist of a variety of hydrocarbons. They are commonly divided into four main groups [3]:

- Saturated acyclic (aliphatic) hydrocarbons (parafins or alkanes)
- Unsaturated hydrocarbons with double bonds (olefins or alkenes) or triple bonds (acetylene or alkynes)
- Acyclic (aliphatic) hydrocarbons, either saturated (cycloalkanes = naphtenes) or unsaturated (cycloalkenes or cycloalkynes)
- Aromatic hydrocarbons (arenes).

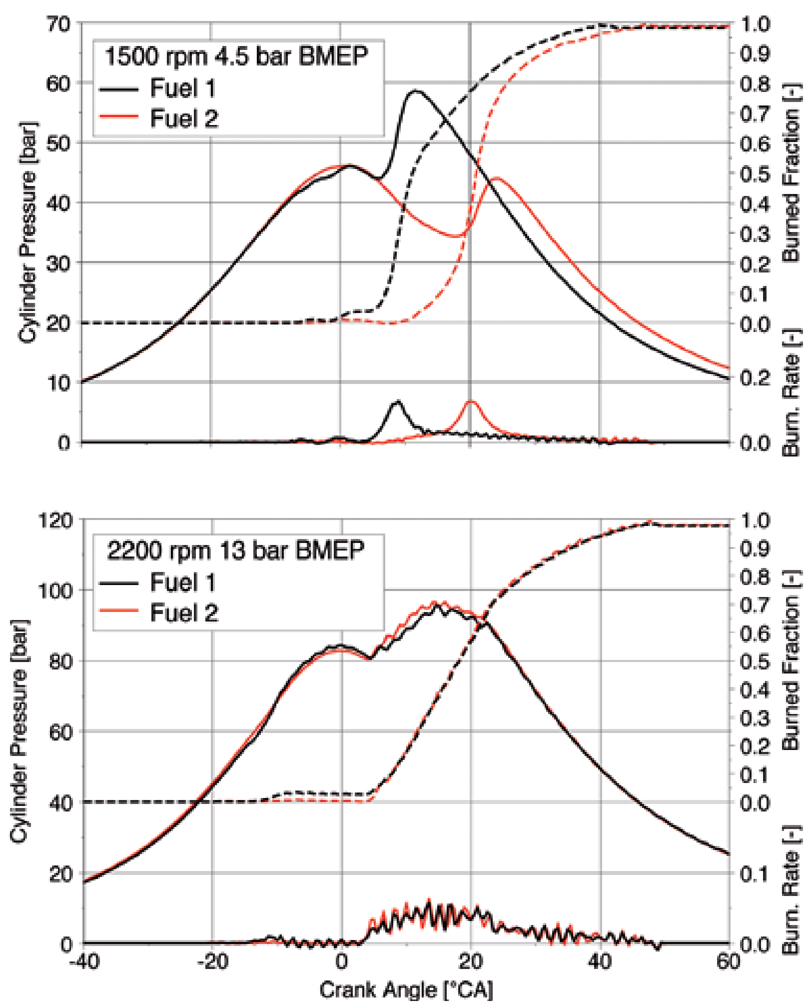

Figure 2: Difference of combustion, investigated fuels

Table 2: Characteristics of investigated fuels

	Fuel 1	Fuel 2
Density at 15 °C [kg/m³]	831.4	852.3
Cetane number	53.6	42.1
Boiling characteristics up to 250 °C [% by volume]	37.4	36,4
Boiling characteristics up to 350 °C [% by volume]	94	>99
Aromatics ratio [% by volume]	25.6	34.5

Table 3: Investigated load points and calibration

	1500 rpm – 4.5 bar PME	2200 rpm – 13 bar PME
Pilot injection quantity [mg/stroke]	1.9	2.2
Pilot injection start [µs]	1700	1400
Main injection start [°CA BTDC]	2.4	2
Rail pressure [bar]	1050	1460
Air-mass flow [mg/stroke]	345	760

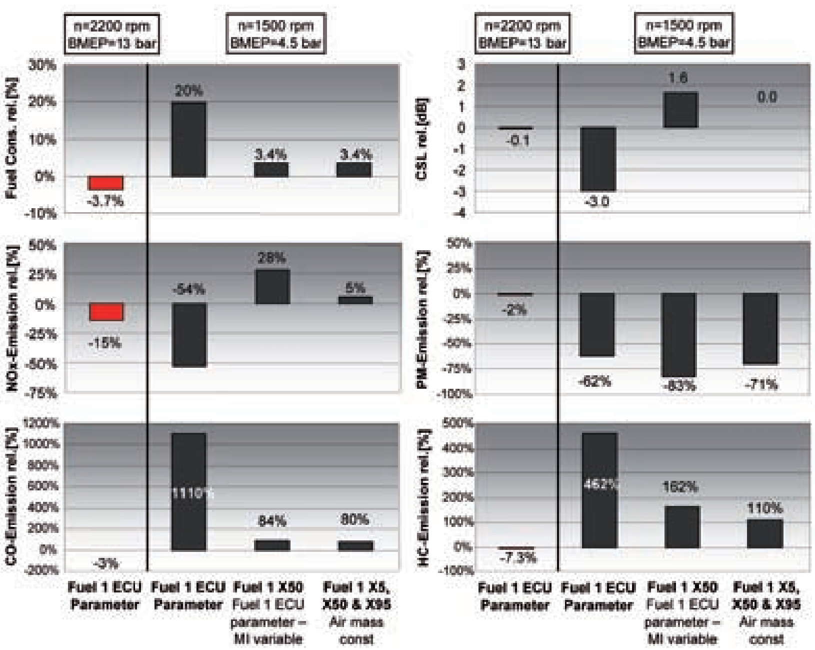

Figure 3: Influence of fuels on emissions in the high and low part load point and compensation measures

Figure 1 shows the boiling temperature and the cetane number for hydrocarbons depending on the number of C atoms. A clear correlation between the number of C atoms and the boiling temperature can be observed. In contrast to this, the cetane number correlates with the number of C atoms only in comparison with the own group. The influence of the structure is substantially more pronounced. In particular with paraffins, the cetane number depends to a large extent on the degree of isomerization. While n-octane with a cetane number of approx. 65 is clearly more prone to ignition than conventional diesel fuel, ISO-octane is the standard for knock-resistant fuels and thus fuel with poor ignition quality (with an octane number of 100 or a cetane number of 12.5). The major influence on the combustion behaviour results from the different binding forces between hydrogen and the carbon atoms. The lower number of primary bonds in unbranched molecules leads to an increased reactivity, since the likelihood that the weak secondary bonds are breaking up earlier is substantially greater [7].

Due to the thermodynamic principles of the combustion process in diesel engines, the fuel that is being used is an important factor in the entire energy transformation process.

As we know, diesel fuel is injected into the cylinder near top dead centre (OT) at the end of the compression phase in engines with compression ignition when the thermodynamic boundary conditions of the cylinder charge (pressure and temperature) are suitable for mixture formation. As soon as the first fuel quantities enter the cylinder, chemical reactions start to occur and continue as long as more fuel quantities are added, while at the same time the remaining liquid fuel and the fuel/air vapour are intermixed. Auto ignition occurs when this mixture is present in a relatively rich air/fuel ratio of approximately 0.25. After this premixed combustion, the remaining fuel is converted in a diffusion flame when the injection period is longer than the ignition delay.

The fuel composition and properties are inextricably linked and, due to the various combinations and dependencies, have a direct impact on the combustion-relavant process parameters as is shown in **Table 1**.

This table illustrates that a description of the fuel with the help of the cetane number alone is problematic. The boiling characteristics in particular have a great impact on mixture formation, and as a result on combustion as well, whereby a low saturated liquid line contributes to a quick mixture formation and thus promotes a reduction in particulate emissions [4], [8]. A low aromatics content is preferable not only because of a possible health threat, but is generally described as an option for reducing HC, CO, and particulate emissions [8].

3 Basic Research

Two fuels with very different properties are used as a basis for the following studies, **Table 2**. Both fuels are available in the

Table 4: Values for variation of calibration settings

Parameter	Start of preinj.	Pilot injection quantity	Start of main inj.	Rail pressure
Unit	**[°CA BTDC]**	**mg/stroke**	**[°CA BTDC]**	**[hpa]**
Base point	15	1.6	-2.5	800000
Preinj. quantity 1	15	3.2	-2.5	800000
Preinj. quantity 2	15	4.8	-2.5	800000
Early preinj.	18	1.6	-2.5	800000
Late preinj.	12	1.6	-2.5	800000
Early main inj.	15	1.6	0	800000
Late main inj.	15	1.6	-5	800000
Reduced rail pressure	15	1.6	-2.5	600000
Increased rail pressure	15	1.6	-2.5	1000000

US and have representative minimum and maximum values in terms of the cetane number. The aromatics content of fuel 2 with a value of 34.5 % is also to be regarded as a maximum (maximum aromatics content as per ASTM [minimum fuel requirements for diesel engines] D 975-05 Grade 2-D S15: 35 %), whereby the 25.6 % value of fuel 1 characterizes the average in Europe.

The results shown in the following were conducted on a modern full size diesel engine with a cylinder displacement of appr. 500 cm^3.

To illustrate the influence of the greatly differing fuels on the combustion as a function of the load point, we will take a look in the following at the example of the part load points n = 1500 rpm, p_{me} = 4.5 bar, as well as n = 2200 rpm, p_{me} = 13 bar. **Table 3** shows the calibration parameters which are identically at first for both fuels and have been determined for fuel 1. This represents a typical Euro V calibration.

The pressure curve and burn rate of the different fuels in the illustrated load points are shown in **Figure 2**. It can be observed, that in particular in the lowest load point, the pressure curve and burn rate are clearly different. Combustion takes place later due to the significantly longer ignition delay. This is caused by the lower ignition performance of fuel 2 which leads to the effect that the pilot injection is not burning, with the consequence of a longer ignition delay, and the burning of the main injection itself is also retarded.

By comparison, the pressure curve and the combustion at high loads are only marginally affected. Here too, the pilot injection is barely converted in spite of an identical pilot injection quantity. Nevertheless, the combustion during main injection as well as the rate of combustion differs only slightly for the investigated fuels, since, at high loads, the boundary conditions in the combustion chamber in terms of pressure and temperature lead to a clearly shorter ignition delay for both fuels [3]. As a consequence, the ignition delay of the main injection is nearly independent from the pilot quantity, and the conversion rates and burn rates are almost the same.

This simple comparison depicts, that the description of fuel characteristic in-

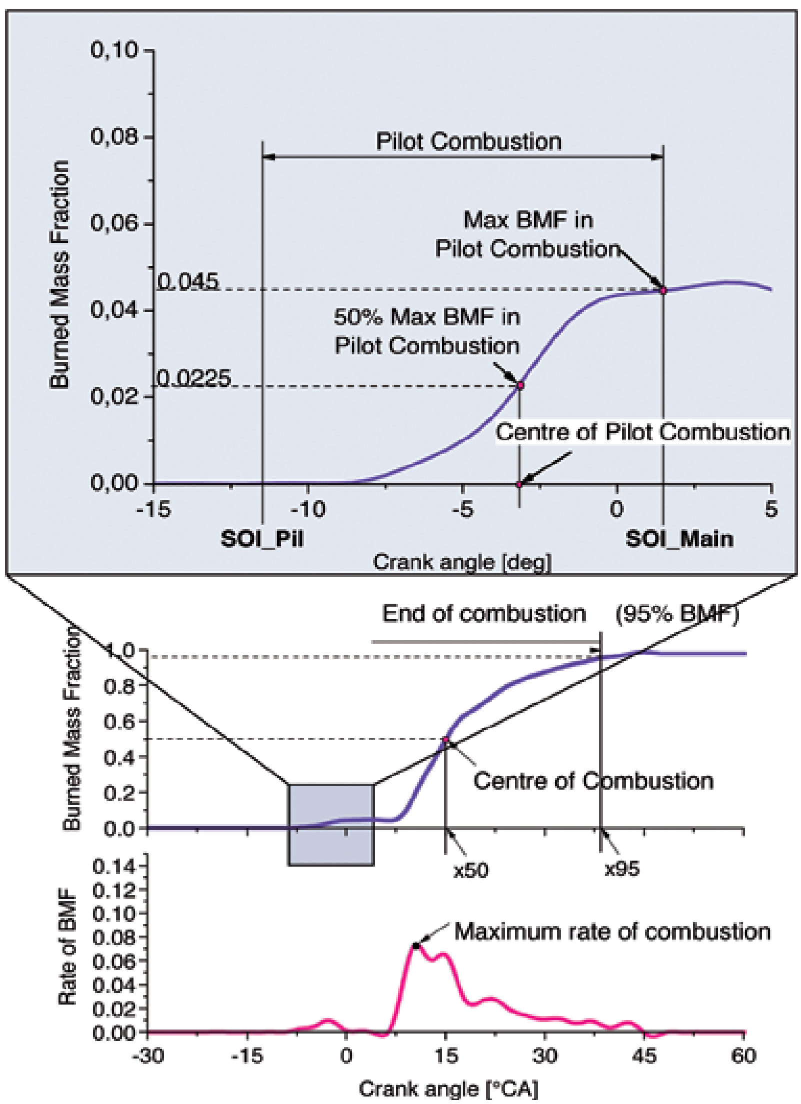

Figure 4: Derivation of combustion-relevant control variables

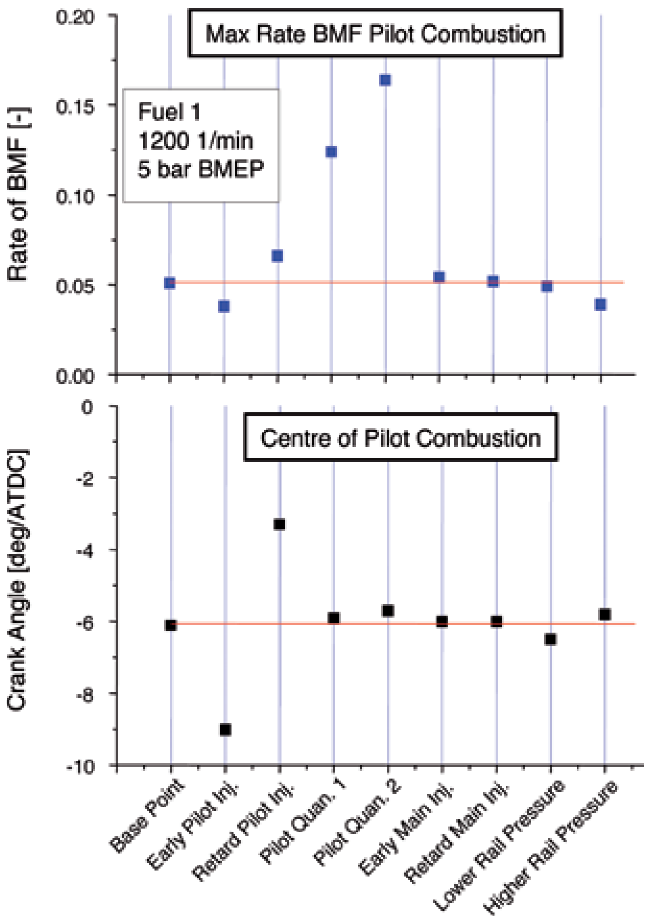

Figure 5: Influence of controlled values on pilot combustion

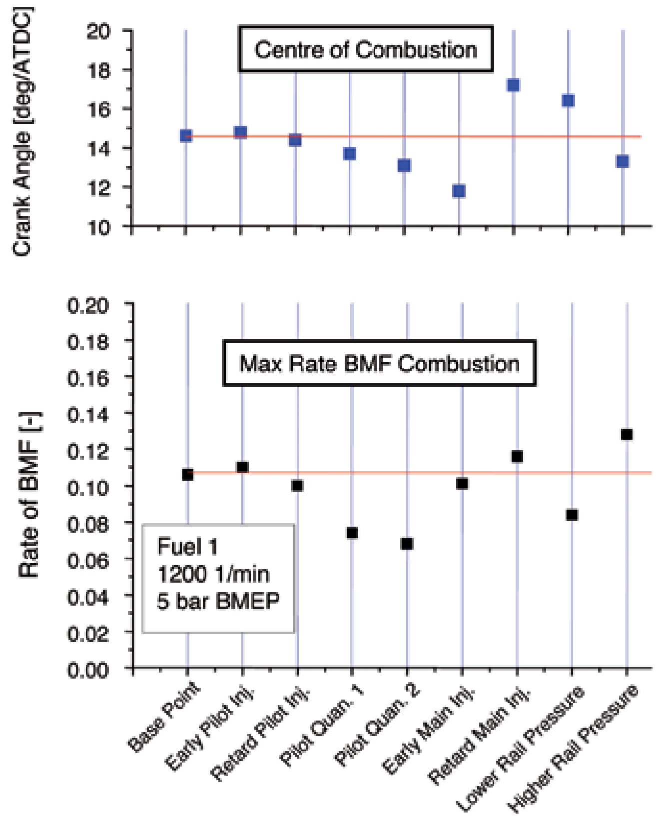

Figure 6: Influence of controlled values on main combustion

teraction with combustion only via cetane number is limited to low load engine conditions, or more general to the pressures and temperatures prevalent in the combustion chamber at low loads. The description using the ignition delay is clearly more meaningful.

Figure 3 illustrates, analogous to the burn rates for the load points that are under consideration here, the influence of fuel 2 with identical calibration on emissions, consumption, and combustion noise. The relevant changes are based on the measurement values determined with fuel 1.

It becomes clear that the different fuel properties in the high load point, similar as with the observed influence on the burn rate, have only a minor influence on the emissions as well as on the combustion noise. By tendency, US fuels rather have a positive influence on the emission level, which is noticeable in particular by a decrease of nitrogen oxide emissions.

The analogy between the course of cylinder pressure and emissions is also clearly noticeable here for the low load point. Consumption increased by 20 % due to the substantially delayed combustion. NO_x and particulate emissions decrease together with the level of combustion noise, whereas the HC and CO emissions are multiplying disproportionally.

These considerations show that an active compensation of the fuel properties must take place predominantly at low loads or in general at poorer ignition conditions.

To verify to what extent the adaptation of the burn rate with fuel 2 at the low load point considered can be used to adjust emissions, consumption, and combustion noise, the calibration parameters main injection start, exhaust gas recirculation rate, pilot quantity and timing, rail pressure and boost pressure were modified by means of DoE. Not only the standard variables such as emissions, consumption, and combustion noise were represented in the resulting models, but also the 5, 50, and 95 % burned mass fraction points of combustion.

In the first step of the secondary evaluation, the specified position of the 50 % burned mass fraction point was set constant. For this purpose, at first the calibration parameters that had been optimized for fuel 1 were set as constant and then the start of main injection was modified in order to achieve the same location of the centre of combustion as for fuel 1.

The resulting emission, noise, and consumption results are illustrated in the third bar in Figure 3. The model prediction is already producing a distinct approximation of the values achieved with fuel 1.

In the next step, only the EGR was fixed in the model and the remaining calibration parameters were changed. After specifying the 5, 50, and 95 % burned mass fraction points as for fuel 1, we obtain the deviations shown in the fourth bar. Combustion noise, HC and

CO emissions, as well as fuel consumption come quite close to the values achieved with fuel 1 with comparable NO_x emissions. The combustion noise in particular and the consumption are nearly identical, the nitrogen oxide emissions can be fine tuned by adjusting the oxygen concentration. The particulate emissions as well as the HC and CO emissions can not be fully adjusted, as they depend to a higher extent on the mixture formation, the post-oxidation processes, and the combustion kinetics. It is not possible to derive these relationships merely with a simple zero-dimensional view of combustion.

Nonetheless, it can be implied that in the engine operation area with a high sensitivity for different fuel characteristics, with identical oxygen concentration in the intake manifold and thus identical nitrogen oxide values, an approximation of the burn rate will at least lead to an adaptation of emissions, consumption, and combustion noise.

4 Combustion Control to Compensate for the Fuel Influence

Based on the results and considerations discussed above, a definition of the characteristic parameters and the control variables of combustion that are used to compensate for the fuel influence will be provided.

4.1 Parameters and Control Variables of Combustion

Figure 4 shows the combustion parameters that are being considered based on a characteristic burn rate with pilot injection. Our first approach is based on the assumption that both the combustion of the pilot and the main injection process should be controlled in order to achieve the most effective compensation possible of the fuel‘s influence, since the conversion of the pilot quantities together with the fuel properties has a big influence on the main injection‘s ignition delay.

However, the enlarged section of the burn rate during the conversion of the pilot injection as shown in Figure 4 can directly be used to derive that it will not be possible to define a fixed value in order to characterize the start of pilot combustion, since the overall burn rate of the pilot can vary greatly depending on the boundary conditions.

In order to obtain a more precise description of the pilot combustion, other parameters are used. The burn rate at the beginning of the main injection process for instance can provide information on the burning rate of pilot combustion. However, it must be noted that the pilot combustion must take place before the start of the main injection. It is not possible to recognize small pilot injection quantities or their combustion during main injection with adequate accuracy due to the influence of the injection as well as its vaporization.

As a result, it is assumed in the following control concept, that the combustion of the pilot quantity is taking place before the start of the main injection.

When we use this as a basis, the combustion start during pilot injection can also be defined as half of the overall burned mass fraction in pilot combustion.

The three parameters that are used to describe main combustion are also illustrated in Figure 4. They are: The center of combustion, characterized by the 50 % burned mass fraction point, the end of combustion, defined as the 95 % burned mass fraction point, and the maximum rate of combustion.

4.2 Analysis of the Control Variables

Suitable control variables must be defined in order to be able to regulate the above defined parameters. For this purpose, the influences of the control variables rail pressure, start of main injection and pilot injection, and pilot quantity on the parameters defined above are shown. The variations listed in **Table 4** were used. The following pictures are based on a series of measurements at the load point n = 1200 rpm, p_{me} = 5 bar.

The influence of the exhaust gas recirculation rate is not considered, because it must be used to control the NO_x emissions and the transient response of the air path is very slow. Therefore the EGR or air mass flow is not suitable as a control variable.

Figure 5 illustrates the influence of the considered controlled variables on the above defined parameters of pilot combustion. It becomes clear that the fuel conversion at the beginning of the main injection process must be adjusted first and foremost through the pilot injection quantity. The rail pressure only plays a very minor role here. The influence at the beginning of pilot injection is also small in this area, since very early pilot injections, which result in combustion failures, must of course be avoided.

As expected, the location of pilot burning centre of combustion is mainly influenced by the start of pilot injection. The other control variables have a negligible impact.

The influences of the control variables under consideration on the combustion process during main injection are clearly more complex. **Figure 6** shows that the start of main injection has the greatest influence on the location of the centre of combustion. However, we can see that the pilot injection quantity also has a major influence on the main combustion position, as it has a dominating impact on the ignition delay during main injection.

Due to the changed injection rates as well as the different atomization and vaporization behaviour, the rail pressure has a significant impact on the combustion position.

It is difficult to directly influence the combustion duration or the end of combustion, the exhaust gas recirculation rate has a major influence on this parameter. However, for the reasons mentioned above, it can not be used to regulate this parameter.

The maximum rate of combustion is also influenced directly by several control variables. The main influence parameters here are the pilot injection quantity as well as the rail pressure. The pilot injection quantity must be ruled out as a control variable for the maximum rate of combustion, though, since it is used to adjust the pilot injection rate. Therefore, it also acts as a disturbance variable for the rate of combustion and can, in an ideal case, be added on together with a decoupler to a possible regulator, which uses the rail pressure as a controlled variable.

4.3 Results

These relationships were assessed one more time and then implemented in a basic control structure as shown in **Figure 7**.

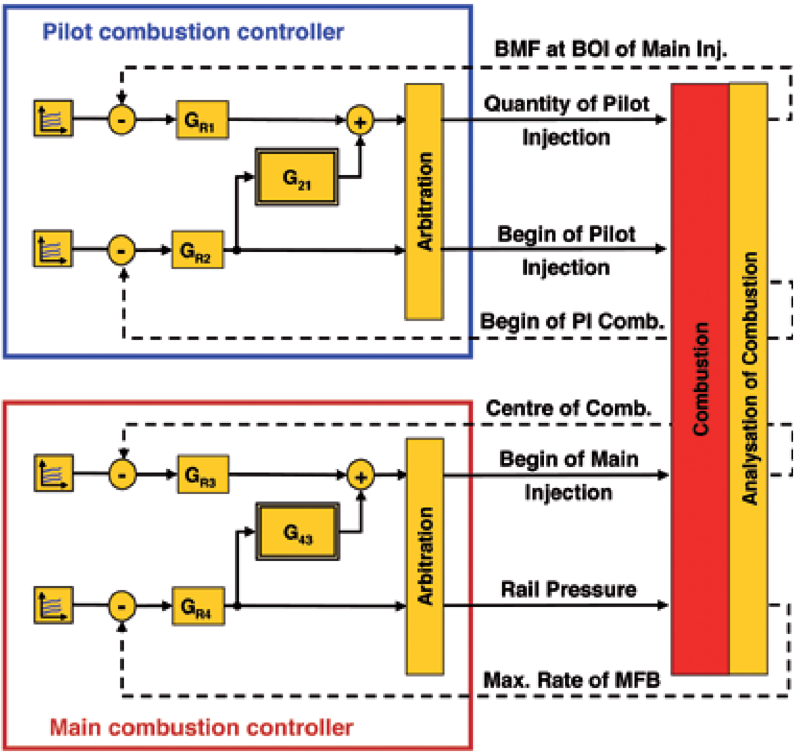

Figure 7: Control structure for fuel compensation

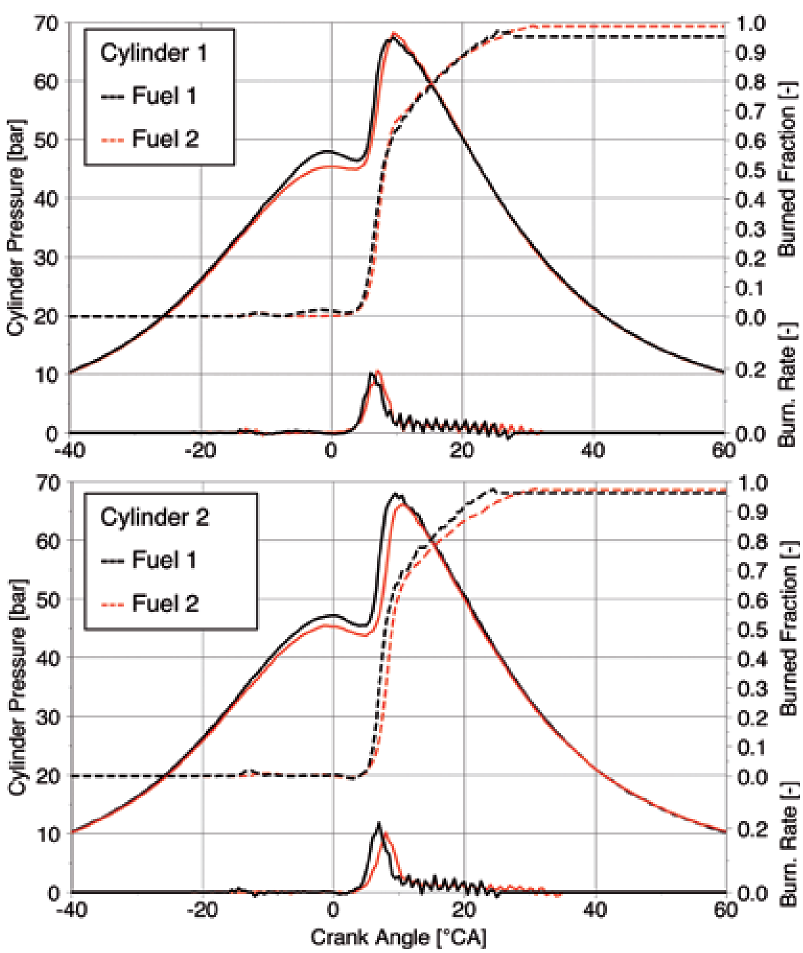

Figure 8: Comparison of combustion for different fuels with activated fuel compensation

The structure comprises a decoupled regulator for the control of combustion during pilot injection. It influences the quantity and the start of pilot injection and makes the adjustment by setting the conversion rate at the beginning of the main injection process as well as the time of half of the conversion of this quantity. This ensures a defined pilot injection and thus constant conditions for mixture formation and inflammation of the main injection quantity. The start of main injection and the rail pressure are the control parameters for regulating the main combustion process. While the start of main injection adjusts the combustion position and thus ensures the most constant combustion efficiency possible, the adjustment of the rail pressure is used to set the maximum combustion conversion rate. The regulators are monitored and in another block for the plausibility of their signals and prioritized. Here various boundary conditions are checked, e.g. the conversion rate during pilot injection.

Some fundamental measurements were performed with this structure at the load point n = 1500 rpm, BMEP = 4.5 bar. By way of example, **Figure 8** shows the pressure curves and burn rates of fuel 1 and 2 for two cylinders of the test engine. In fuel 2, the described combustion regulation is switched on. The location of the centre of combustion and the maximum rate of combustion can be adjusted very well in cylinder 1. The burn rate of the pilot quantity for fuel 2 is substantially lower. Here it can be seen, that it is difficult to regulate the very low values of the rate of combustion with the pilot injection. The burn rates of the second cylinder are not as congruent as for cylinder 1. But in comparison to Figure 2, where the fuels are compared without any compensation measures, a clearly improved combustion congruence can be achieved. There is no significant conversion of the pilot quantity in cylinder 2 for both fuels.

As illustrated in **Figure 9**, the emission and noise behaviour come substantially closer to the results with the fuel 1 in the load point under consideration when the fuel compensation is activated. The measurements were performed with a constant air mass flow.

Only the CO emissions are significantly higher, which is due to the lower rate

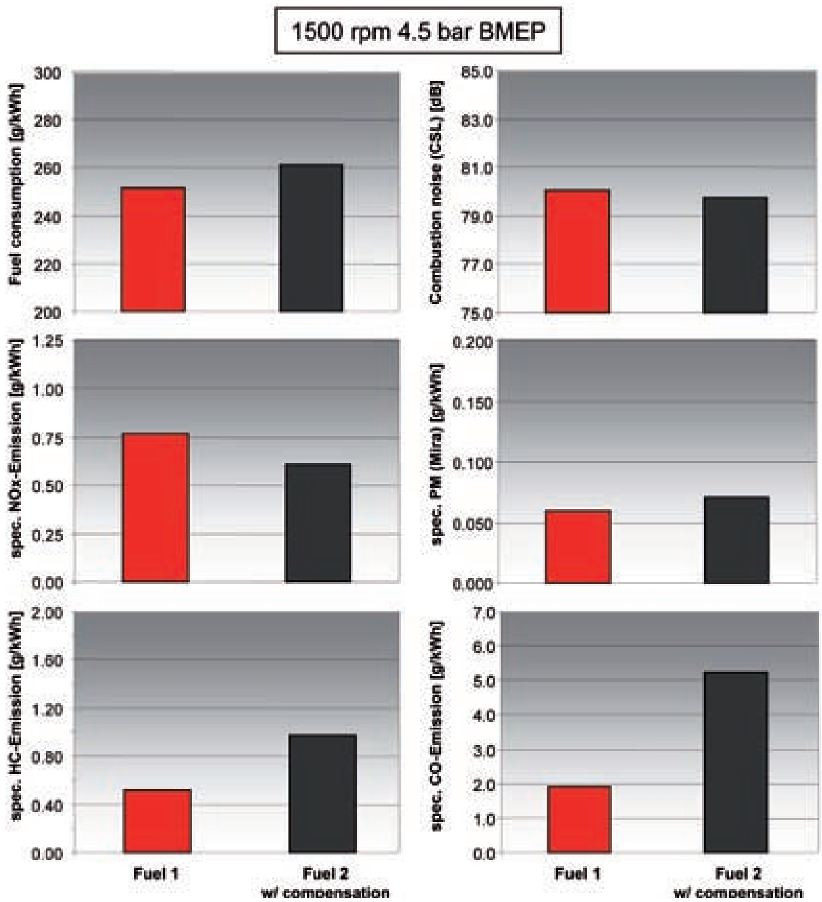

Figure 9: Emissions at a low part load point with activated fuel compensation

of combustion of the pilot quantity. However, this in turn emphasizes the importance of the strategy of also including the rate of combustion of the pilot quantity into the control algorithm. As a matter of principle, we also have to consider that the basic calibration that was used as a basis here is geared towards European applications and the pilot quantity is too low for applications with widely differing fuel characteristics. It can therefore be concluded that, for applications such as fuels in the US for instance, higher pilot quantities will have to be used in order to generally reduce the sensitivity on the one hand and to improve the controllability of the pilot quantity burn rate on the other hand.

5 Summary and Outlook

Assuming an expansion of diesel engines into markets with widely differing fuel grades as well behind the backdrop of the introduction of bio fuels, the demands on engine calibration, basic combustion systems, and combustion regulation will increase. In this article, FEV's approach for compensating different fuel characteristics during combustion is illustrated.

Using the example of two fuels with clearly different specifications, we discussed the influence on combustion without any corrective measures at different load points. We were able to demonstrate that, in engine map areas with a high sensitivity for different fuel properties, the adjustment of the combustion characteristics can be used at least in part to compensate for the fuel's influence. Moreover, we discussed basic factors that have an influence on combustion and derived the resulting correlation of the regulation and control variables. The control concept we devised allowed us to adjust the combustion characteristics in steady-state operation and thus in part compensate the influence of different fuel characteristics.

However, even with the approach of an intelligent combustion regulation in order to compensate the different fuel characteristics, the limitation of the fuel variances has to be considered, because a full compensation is not possible. With smaller fuel variances also the emission and consumption variances can be kept very small.

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