# Advanced Diesel Combustion

## A Method Demonstrating Favorable Untreated Engine Emissions with Improved Consumption Characteristics

Future emission norms will further reduce the vehicle emissions of diesel engines. To meet the goal of achieving these stringent limits while maintaining attractive attributes of marketability, the combustion system needs also to be revised or redesigned in order to achieve the $CO_2$ limits that will be demanded in the future, while simultaneously producing significantly reduced untreated $NO_x$ emissions. In the scope of this article, the basic conditions for the design of future combustion systems shall be discussed. As a possible solution, we will introduce the FEV combustion system HECS (High Efficiency Combustion System), with which very low fuel consumption can be obtained even at high part loads, while producing minimal nitrogen oxides.

## 1 Introduction

Since the beginning of the 90s, high-speed direct injection diesel engines have made significant strides in the West European passenger car market. Apart from the traditional fuel consumption advantage during on road driving, the primary factors for this success have been a substantial improvement in performance and power density as well as a massive progress in the acoustic properties.

Future emission norms (EU-6) and the legislation in other markets (US Tier2 Bin5) will significantly reduce the permissible vehicle emissions of diesel engines even further. If there is no major change to the combustion system or the combustion relevant hardware (injection system, boosting, exhaust gas recirculation system, gas exchange, flow characteristic in the combustion chamber), then any further reduction in nitrogen oxides, either raw or after treated, will lead to a significant deterioration in the fuel consumption [1]. This is counterproductive with respect to the limitation of the $CO_2$ emissions being required in the future. Downsizing methods also necessitate lowering nitrogen oxide emissions in the high load range with an acceptable or possibly even improved specific consumption. This is due to the fact that the emissions-relevant load range is shifted towards high part loads through downsizing. At the same time, the attractive attributes offered by today's diesel engine, such as performance, low fuel consumption and acoustic properties on par with those of gasoline engines must be retained.

Therefore, the combustion system must be developed for emission regulations beyond EU-5 or those in force in other markets in order to resolve the conflict between the different target values, while maintaining or even improving the fuel consumption advantage innate to the diesel engine.

Hence, the framework for the design of future combustion systems shall be discussed in this article. As a possible solution, we will introduce the FEV combustion system HECS (High Efficiency Combustion System), with which a very good fuel consumption and low nitrogen oxide emissions can be achieved. The combustion system was named to reflect the good fuel consumption.

## 2 Basic Diesel Engine Combustion Methods

The general and overall design of a diesel engine combustion system usually includes the following setting parameters:
- Fuel injection with nozzle geometry (number of holes, hole diameter, hole arrangement, cone angle, needle seat), injection pressure and characteristic injection timing, injection strategy and injection profile (ramp, rectangle, „boot injection", free control of the shape)
- Combustion chamber with bowl geometry, compression ratio and dead space
- Cylinder charge with thermodynamic state variable, composition of the filling and flow condition (charge movement)
- Gas exchange.

The general goal of combustion system development with optimised combustion, lowest possible emissions and good fuel consumption has always been to minimize soot formation or to intensify soot oxidation. At the same time, the formation of $NO_x$ is suppressed through an evenly distributed combustion with a low oxygen concentration and thus lower peak combustion temperatures.

Since the variables listed above interact with each other, the design of a new and advanced diesel combustion system is characterised by a close meshing of the application of CAE techniques and experimental work. This needs to be done based on a broad portfolio of experience and a representative database of previously implemented combustion systems, also taking special applications (e.g. motor racing) into account.

Based on the requirements in the performance specifications that must be met (performance, fuel consumption and emission targets) and depending on the possibly already given base specifications of the engine, the combustion system is predefined, **Figure 1**. This is done by linking the benchmark with pre-liminary piston bowl design and the injection hardware by means of CFD and a detailed definition of the flow characteristics as well

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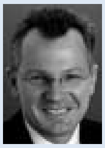
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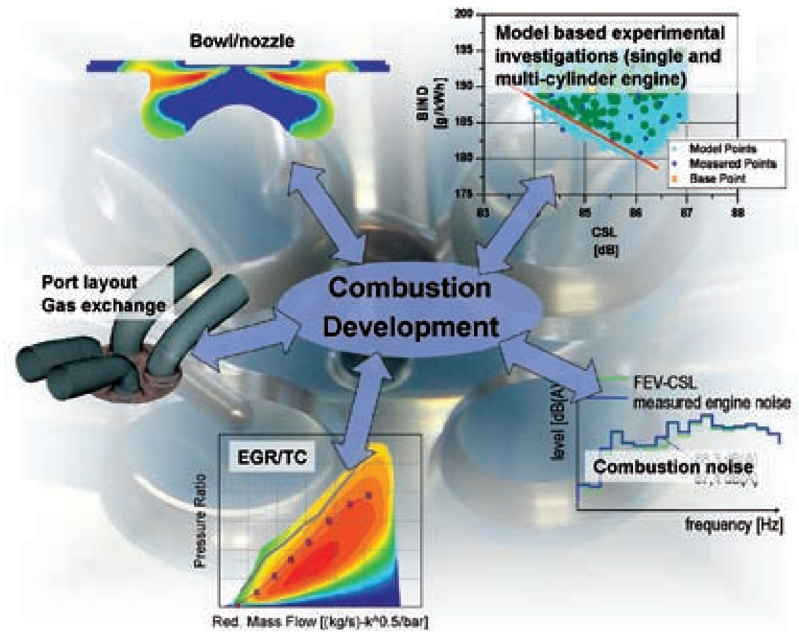


**Figure 1:** Tools and optimization parameters of the HECS combustion system development

as that of the EGR and boosting system. Through experimental verification and with the methods and DoE approaches established at FEV (including those for a single cylinder engine), it is possible to optimise all parameters of the combustion system in a short period of time by continuously and iteratively using CFD analysis. All the optimisation steps are accompanied by a combustion noise analysis.

This optimisation process was also used to make the combustion system shown here ready for application.

## 3 Definition of the Goal for a Fuel Consumption Optimised Combustion System

### 3.1 Theoretically Achievable Limitation Cycle

For a conventional reciprocating piston engine, the constant-volume cycle is the cycle with the best efficiency and is thus considered the thermodynamic limit for the efficiency of the high-pressure cycle. The efficiency is given by: $\eta_{th,v} = 1-\frac{1}{\varepsilon^{\kappa-1}}$ [3]. Consequently, the process loss of the constant-volume cycle depends solely on the compression ratio and the material properties of the working fluid. By comparison, the efficiency of the Seiliger cycle is a function of the permissible peak pressure, the pressure at the start of compression, and the supplied heat. The efficiency, particularly that of the peak pressure-limited Seiliger cycle, improves only moderately with increasing compression ratio in the feasible range for diesel engines. In contrast, the dependence of the cycle efficiency on the adiabatic exponent is significantly more pronounced.

The gas properties of the real working fluids are a function of the temperature and the composition, which in turn depends on the combustion air-fuel ratio and the exhaust gas recirculation rate. The tri-atomic molecules $CO_2$ and $H_2O$, which are unavoidable as the end product of a complete combustion, drastically reduce the adiabatic exponent. Primarily, only the combustion air-fuel ratio is of importance for the exhaust gas composition or the composition directly after combustion. $CO_2$ and $H_2O$, which are introduced through the exhaust gas recirculation, are of relevance for the compression and combustion begin. This is the main reason for requiring as much excess air as possible, as the basis for high cycle efficiency.

The specific heat supplied has the biggest influence on the peak temperature. For a given constant energy supply, the specific heat supplied increases linearly with the specific volume at the start of compression. An increase in filling thus lowers the peak temperature and thus the cycle losses. The increase in density can occur through an increase in pressure and a decrease in filling temperature. In the process as a whole, the temperature decrease has a dual impact on the peak temperature, as the temperature level of the entire cycle is lowered in addition to the reduced specific energy supply.

The formation of thermal nitrogen oxides increases exponentially with the combustion peak temperature in a locally stoichiometric mixture. Though the peak temperature depends on the temperature before combustion, it is mostly governed by the oxygen content of the fresh mixture. **Figure 2** shows the trace of nitrogen oxide concentrations in the exhaust gas plotted against oxygen concentration in the intake manifold for an EGR variation experiment at eight different load points. We can see a high correlation, even though the injection process and the filling temperature vary.

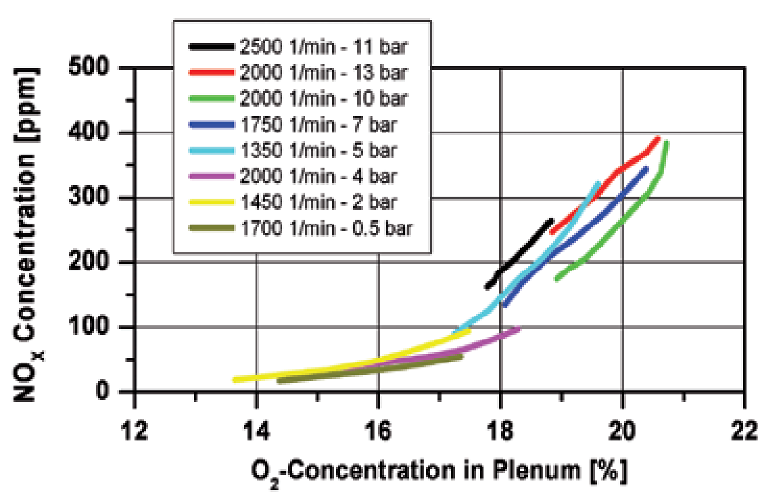


**Figure 2:** Dependence of the nitrogen oxide concentration on the oxygen concentration in the plenum of various, emission-relevant load points

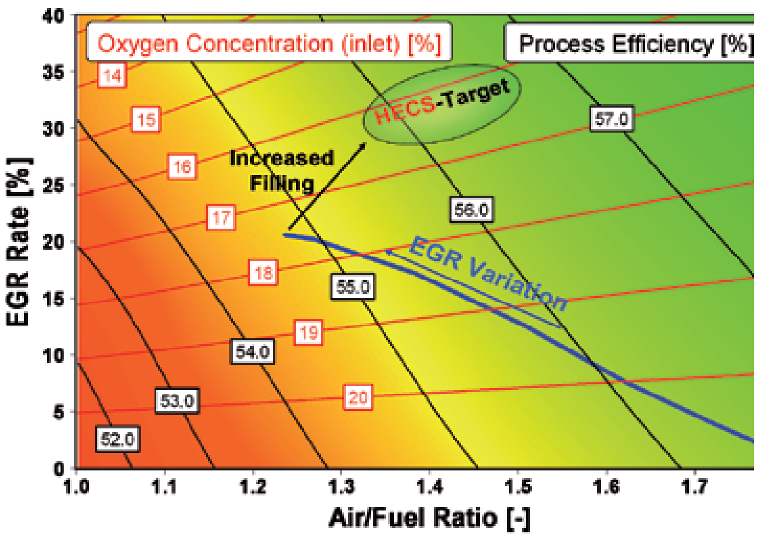


**Figure 3:** Dependence of the process efficiency on the different exhaust gas recirculation rates and combustion air-fuel ratios – 30 mg of injected fuel

The seemingly conflicting demand for a low oxygen content prior to combustion for a nitrogen oxide reduction and simultaneously high air excess in order to increase the efficiency and lower the soot content can be solved through improved filling. In the process, lowering the temperature before the compression phase seems to be the more effective measure from a thermodynamic point of view.

Overall, the theoretical consideration of the air standard cycle requires a very good and cold filling of the combustion chamber, in order to substantially improve the nitrogen oxide/ fuel consumption trade-off. **Figure 3** shows the efficiency of the isochoric cycle as a function of the combustion air-fuel ratio and the exhaust gas recirculation rate given the following boundary conditions:
– 30 mg of diesel is burnt isochorically in real gas
– No burnt gas remains in the combustion chamber and the gas exchange takes place without any losses
– The temperature before the start of compression is 323 K (constant)
– Compression ratio 15.

The results of the cycle calculations show that the cycle efficiency improves with an increasing air excess and with an increasing exhaust gas recirculation rate. Also shown is the trace of the exhaust gas recirculation variation of an EU V combustion system at a load point with a similar injection quantity. A further reduction in the oxygen concentration to approximately 16 to 17 %, to meet the EU-6 limits, by increasing the exhaust gas recirculation rate would lead to an excessive reduction of the combustion air-fuel ratio. The corresponding negative influence on the process efficiency must be prevented, even if it were possible to minimize soot formation through an increase in homogenization. The target of the HECS system that is defined accordingly is also shown in Figure 3. Both the exhaust gas recirculation rate and the air-fuel ratio can be increased through increased filling in order to lower the nitrogen oxide emissions to the EU-6 level and at the same time create the prerequisites for high cycle efficiency through reduced cycle losses. Besides, it is possible to avoid the soot problem through an increased air excess, thereby, promoting fast fuel burning rates.

## 4 Consumption Potential of a Conventional Combustion System

After the theoretical consideration of achievable cycle efficiency and definition of the boundary conditions of future combustion systems with respect to oxygen concentration and combustion air-fuel ratio in the high load range, the practically achievable diesel engine fuel consumption shall be discussed.

To maximise the fuel consumption potential of a single stage, turbocharged EU V combustion system with state of the art injection system, exhaust gas recirculation cooling, compression ratio and charge motion, the calibration was adapted at the test bench at two load points without taking emissions and acoustics into account. **Figure 4** shows the break-up of losses in each case for the base point and the optimum point of optimum fuel consumption. At the lowest load point (1500 rpm, BMEP = 3 bar), it was possible to realise a minor consumption reduction, which, at this load point, is mainly the result of the reduced cycle losses, made possible through an increase in the combustion air-fuel ratio from 1.8 to 3. Although the displacement of the centre of combustion in the direction of top dead centre reduces the non-ideal com-bustion losses, the higher temperatures lead to an increase in cooling losses.

In the high-load operating point (2000 rpm, BMEP = 12 bar), we can see a significantly increased potential. The specific fuel consumption can only be lowered from 233 g/kWh to 209 g/kWh by changing the calibration. It was possible to reduce the cycle loss from 45.5 to 44.1 % by increasing air excess from $\lambda$ = 1.3 to 1.7. The higher air excess implicitly also has a positive effect on the losses occurring during real and incomplete combustion, since the combustion is a lot faster. Furthermore, the centre of combustion was brought forward by advancing the injection timing from 13° to 6.5° after top dead centre. The 3.1 % reduction in real and incomplete combustion losses offset an increase in cooling loss of 1.7 %.

The cycle losses are expelled as heat in the exhaust gas for the most part and can at least be used in part in the turbine of the turbocharger. This, theoretically, results in the potential for generating a positive gas exchange loop. In this case, given sufficiently high exhaust gas enthalpy and a good turbocharger efficiency, it is possible to utilise the energy at the piston during gas exchange, which otherwise would have been written off. However, in conventional high pressure exhaust gas recirculation systems, very high pressure ratios are sometimes necessary in the exhaust gas recirculation tract in order to establish high EGR mass flow rates through the cooler and the across the valve. These scavenging pressure drops that are frequently caused by VTG and/or throttle elements have an extremely negative impact on gas exchange. In the selected high load point, the gas exchange loss was reduced from 2.7 % to 1.5 % by dethrottling the VTG and increasing the exhaust gas mass flow rate, even though the exhaust gas enthalpy was significantly reduced.

The improvement in fuel consumption of 10 % in the high load point can therefore be attributed in equal measure to the improved process cycle, the earlier and faster combustion, and the lower gas exchange losses. The goal of the FEV HECS combustion system's design is to realise the fuel consumption potential already shown through the effective optimisation of the boosting process and the combustion chamber geometry, while at the same time lowering the engine's untreated nitrogen oxide emissions to conform to future emission limits.

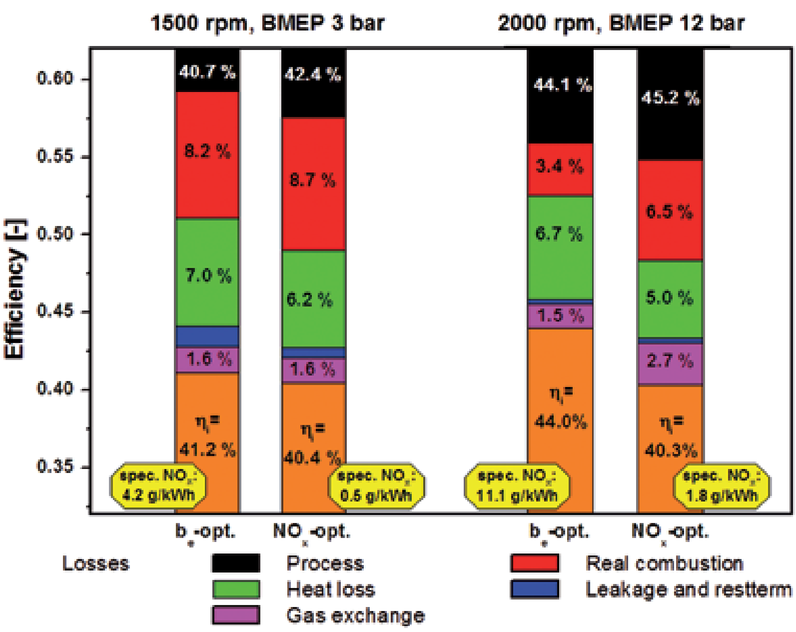


**Figure 4:** Break-up of losses in the part load points 1500 rpm, $p_{me}$=3 bar and 2000 rpm, $p_{me}$=12 bar, in each case with calibrated for minimum nitrogen oxides emissions and minimum fuel consumption

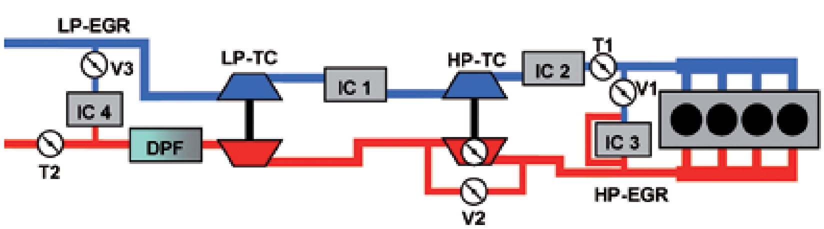


**Figure 5:** Schematic of the boosting and exhaust gas recirculation system that is used to meet the HECS target range for the exhaust gas recirculation rate and combustion air-fuel ratio at high loads

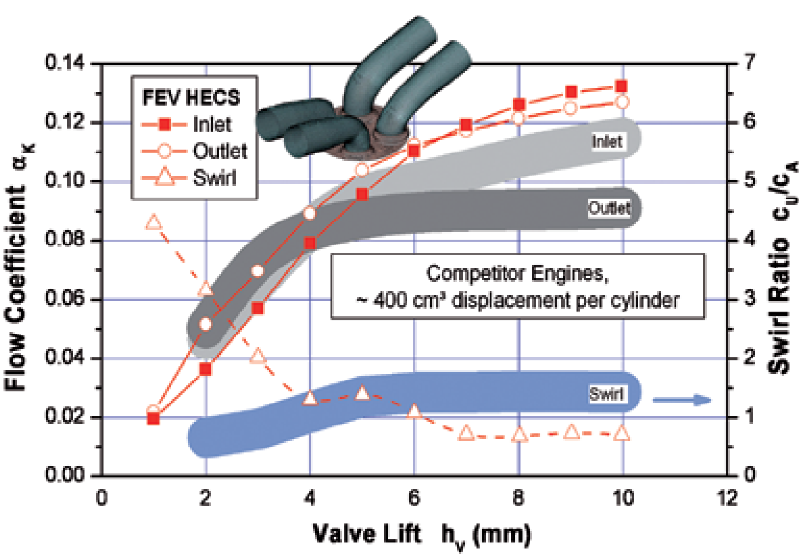


**Figure 6:** HECS flow characteristic and comparison to conventional processes

## 5 Definition and Potential of the HECS Combustion System

### 5.1 Hardware for the HECS System

It was demonstrated in the previous sections that the cylinder filling and an optimised gas exchange are the key factors needed to realise excellent fuel consumption with low nitrogen oxide emissions. To achieve the goal of obtaining good cylinder filling quantity and low gas exchange losses, the boosting, the exhaust gas recirculation, and the port concepts were completely revised for the HECS combustion system.

The FEV HECS combustion system features a 2-stage boosting system with intercooling, **Figure 5**. The two charge air coolers IC1 and IC2, Figure 5, are designed as air/water coolers with a second low-temperature circuit in order to achieve maximum cooling power at small air flow rates. To regulate the boost pressure, the small high-pressure turbine is equipped with variable guide vane geometry (VTG) and a by-pass.

The exhaust gas recirculation system features both a high-pressure and a low-pressure path. The cooler of the high-pressure path is supplied with coolant from the engine circuit. The cooler of the low-pressure path is fed from a low-temperature circuit. The whole low-temperature circuit tract is optimised for a high flow rate, so that the exhaust gas throttle (T2) only has to be used at a minimum level. The control concept of the air and exhaust gas recirculation tracts is based on a model-based approach, which can continuously adjust the most favourable combination of low-pressure and high-pressure EGR, with the help of the two exhaust gas recirculation valves (V1, V3) and the exhaust gas throttle (T2). This allows the two turbochargers to be operated in the region of best possible efficiency.

During the port design, particular emphasis was placed on optimised flow properties, as shown in **Figure 6**. This not only applies to the flow quality, but also to the robustness of the swirl generation in terms of the variance both between the individual cylinders and the different cylinder heads during the production process.

One intake port is designed as a filling port and the other as a classical tangential port. The generation of the charge motion is supported by seat swirl chambers on both intake valves. With the help of this optimised design, it was possible to significantly increase the flow rates, particularly those of the outlet port. This also has a positive effect on cylinder filling, the residual exhaust gas flow rate and the flow losses. Therefore, it constitutes an important component for the improvement in the consumption/nitrogen oxide trade-off.

The layout of the combustion chamber geometry consists of a conventional piston bowl, which, together with the nozzle geometry (8-hole nozzle, ks=1.5) was further optimised in order to achieve very good air utilization. The recessed valves make it possible to eliminate valve pockets in the piston and thus further improve the flow quality in the vicinity of the bowl. A compression ratio of 15:1 was used to achieve an acceptable peak pressure in spite of the increased charge density. The system achieved a specific output of more than 80 kW/l at maximum combustion peak pressures of below 180 bar. Thus, a 1.6 l engine could achieve the performance of a single-stage turbocharged 2.0 l engine. Further, due to the down-sizing effect, this leads to an additional fuel consumption advantage [2]. The injection system used is an advanced common rail system with quick injection needle opening and closing at a maximum injection pressure of 2000 bar.

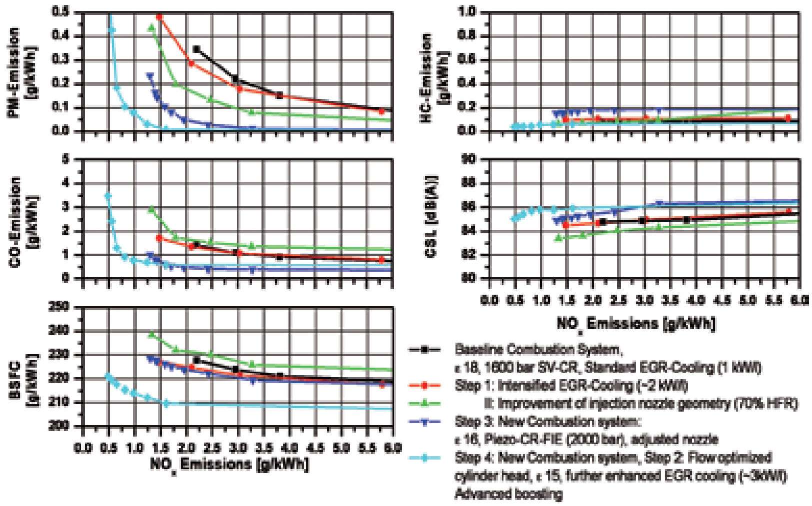


**Figure 7:** Evolution of the HECS combustion system at a high partial load

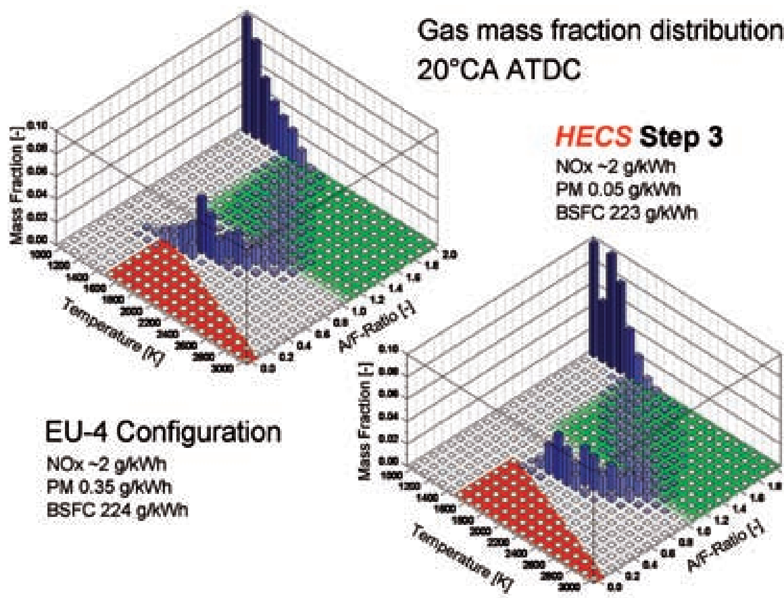


**Figure 8:** λ/T classification of the mass fractions in the combustion chamber at 20°CA after top dead centre in the Euro 4 standard and HECS combustion system, high part load

## 5.2 Potential and Development

**Figure 7** shows, exemplarily, an overview of the development stages starting from a representative EU-4 combustion system to a HECS combustion system for a high-load operating point. Starting with an unmodified base engine, the parameters exhaust gas recirculation cooling and nozzle definition were analyzed. Using enhanced exhaust gas recirculation cooling leads to an improvement in the $NO_x$ PM trade-offs by approximately 10 % due to a higher air excess, in particular at higher exhaust gas recirculation rates. Due to the increased air excess with enhanced exhaust gas recirculation cooling, the improvement of the $NO_x$ PM trade-off can be achieved, while maintaining fuel consumption unchanged. Negative impacts of an optimised exhaust gas recirculation cooling on HC and CO emissions as well as combustion noise generation can not be detected.

The transition to an adjusted nozzle configuration with smaller nozzle holes shows a major improvement of about 40 % in $NO_x$ PM trade-off. The reason for this is the improved air utilisation due to an optimised mixture formation with significantly decreased particulate emissions for a given exhaust gas recirculation rate. However, reducing the nozzle hole diameter leads to longer injection and combustion durations at higher loads, which, in combination with a reduced air excess and the now increased exhaust gas recirculation rates, lead to a deterioration in the efficiency. Hence, for a given ratio of $NO_x$ to PM emissions, the result of the smaller nozzle holes is a deterioration in consumption by 2.5 %.

The optimisation steps described above were integrated into the first development stage of the HECS combustion system. To utilise the advantage of an enhanced mixture formation with lower in-

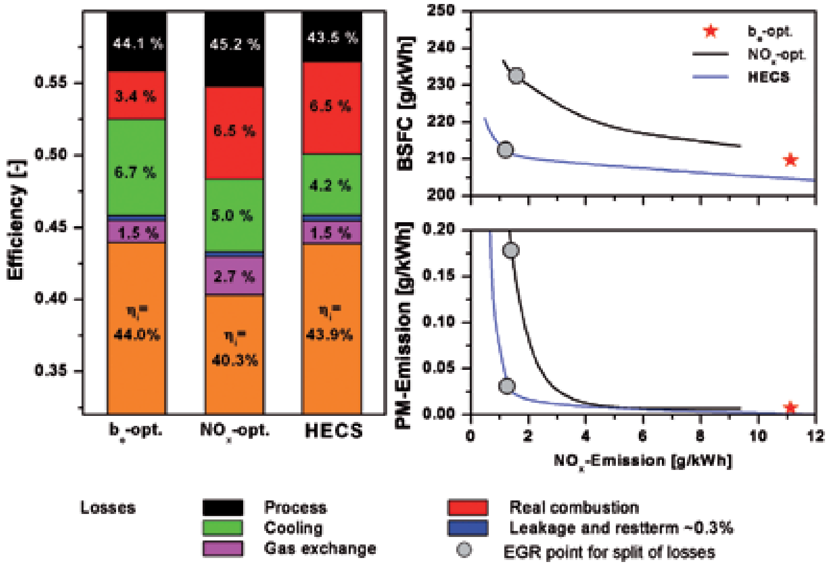

**Figure 9:** Break-up of losses and trade-offs at the part load point 2000 rpm, $p_{me}$=12 bar

jection orifice diameters without having any fuel consumption drawback, an injection nozzle with eight nozzle holes instead of six was used. Furthermore, the compression ratio was lowered by two units in order to obtain a favourable combustion chamber geometry with lower free spray lengths as well as improved full-load characteristics. With these measures, a reduction in the $NO_x$ PM trade-off of approximately 50 % is achievable; fuel consumption as well as HC, CO, and the acoustic behaviour remain comparable.

**Figure 8** illustrates an example of the improvement in mixture formation. Here, the λ/T gas mass distribution in the combustion chamber for the EU-4 configuration and the HECS system are shown at a crank angle of 20° after top dead centre.

Especially, in the region between 0.6<λ<1, the significantly higher homogeneous gas mass distribution in the HECS system, can be seen. Moreover, with the HECS system, there are no mass components in the λ-T-range (marked in red) that are relevant for soot formation, whereas the distribution in the region marked in green, representative of the conditions for a good post oxidation of the soot, is similar.

The boosting system was left unmodified compared to the EU-4 standard variant in the development steps discussed so far. Thus, the use of a powerful, 2-stage boosting system as well as the further development of the combustion system form the next optimization steps of the HECS concept. The design and the development of the boosting group are described in [2]. The further development combustion system mainly consists of a flow-optimized cylinder head and a further reduction of the compression ratio to a ratio of 15:1.

With this fine-tuned technology package, it is possible to significantly reduce untreated emissions even further while improving fuel consumption values. Compared to the EU-4 base, a 70 % improvement in $NO_x$ PM trade-offs was achieved. In contrast to the concepts considered thus far, this $NO_x$ reduction comes along with a fuel consumption advantage of 5 %. The next section will provide a detailed analysis of this improvement in efficiency.

### 5.3 Differentiated Analysis of the Efficiency Improvement

In **Figure 9** the break-up of losses of the HECS combustion system for the high load point 2400 rpm, $p_{me}$=13.5 bar was appended. It also shows the $NO_x$-particulate and fuel consumption-$NO_x$ trade-offs.

The effective optimisation of the air path for maximum cylinder filling and cooling makes it possible to increase both the exhaust gas recirculation rate and the air/fuel ratio. As a result, the process temperature drops to the point where cooling losses are significantly lower, and where the process losses are even lower than in the „EGR-free" variant optimised for fuel consumption. The two-stage charging process and the use of low pressure exhaust gas recirculation systems in combination with the ports result in gas exchange loss levels, which are comparable to those of the calibration optimised for fuel consumption, in spite of the considerably larger filling quantity.

## 6 Summary and Outlook

In the scope of this article, the technical potential of an advanced diesel engine combustion system designed was discussed, which, in addition to achieving very low untreated engine emissions to meet the most stringent exhaust emissions standard, also retains the traditionally good efficiencies of diesel engines. The key elements for achieving the demonstrated potential are:

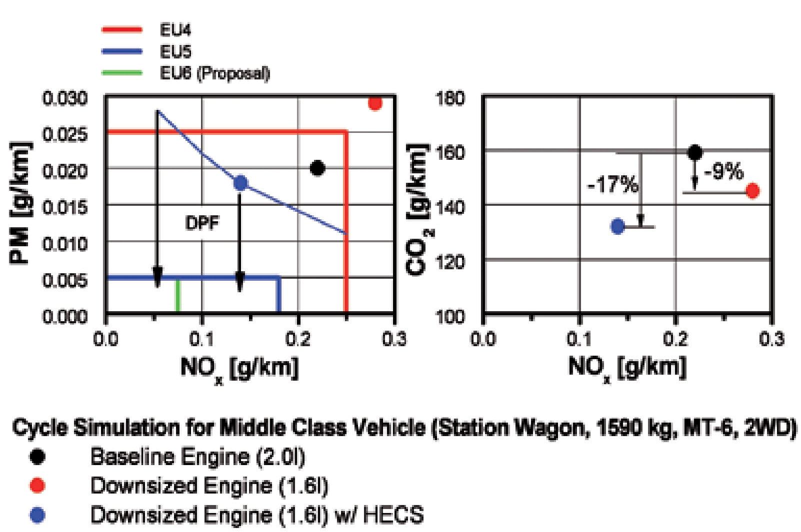

**Figure 10:** Emission and $CO_2$ potential of HECS in the NEDC, vehicle with a flywheel mass classification of 1590 kg

- Charge motion and gas exchange
- Cylinder charge cooling (fresh air and exhaust gas recirculation)
- Fuel injection and mixture formation.

The intelligent combination of these aspects with the help of advanced development tools (CAE, DoE) provides further options for lowering the vehicle-based $CO_2$ emissions, since smaller displacement volumes can be controlled as a result of the reduction of the displacement („Downsizing"), while at the same time the power density is further increased. **Figure 10** shows the potential of a representative vehicle with a flywheel mass classification of 1590 kg, based on cycle simulations of stationary engine results. Starting with a 2.0 l base engine with an EU-4 base system, the EU-4 limits could, initially, no longer achieved with a displacement reduction to 1.6 l; the improvement in fuel consumption is approximately 9 percent. If the characteristics of the HECS system described above are applied to this engine, it is possible to achieve a fuel consumption reduction of approximately 17 percent while meeting the EU-5 limits. The EU-6 $NO_x$ limits can also be met, although with a reduced fuel consumption advantage. Nevertheless, the $CO_2$ emissions would still be clearly below those of the 2.0 l EU-4 baseline engine.

In the future, this will lead to further market options for passenger car diesel engines as powertrains that are environmentally friendly with fun to drive.

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