The CCS Combustion System from Volkswagen

Due to changing economic, technical and social environments and resulting customer demands, future vehicle drive systems will be required to have lower pollutant emissions and greater efficiency, leading to a CO_2 reduction. Combining the advantages of the petrol and diesel combustion processes is a way of achieving these objectives. In the Group Research – Drive Systems of Volkswagen AG, this implementation is referred to as the CCS Combustion System.

1 Motivation

Current engine developments show a trend of increasing similarity between the petrol engine and the diesel engine, **Figure 1**. One promising approach is the partially compression-ignition petrol engine (GCI) [1]. Another one is the partially homogeneous compression-ignition combustion process based on the diesel engine. The final result of the combination of the two combustion processes is referred to as the CCS Combustion System. This combustion process reduces emissions of the pollutants NO_x and particulates, which are subject to legal limits, while significantly increasing efficiency at the same time. Research into its implementation is based on diesel and petrol engines [2].

NO_x and particulate matter emissions can be reduced by early compression ignition of a homogeneous mixture, brought about by early injection and extremely high exhaust gas recirculation rates (EGR). With the example of a diesel engine, **Figure 2** shows that when EGR is continuously increased, the initial rise in particulate matter emissions is subsequently fully eliminated while NO_x emissions continue to decrease. This is an effect of the homogenisation, which prevents rich fuel zones of low air concentration during the combustion process. In addition, the effect of ballast substance in the form of recirculated exhaust gases can be used to keep the 50-% energy conversion position within the optimum range, thus producing a significant increase in efficiency compared with the conventional late peak combustion pressure.

However, as load increases, it becomes more and more difficult to maintain this combustion process. In a diesel engine, it is the insufficient time remaining for homogenisation as the injection quantity increases that necessitates the changeover to the traditional diesel combustion process. The petrol engine, on the other hand, suffers from an inability to deliver the required quantities of recirculated exhaust gas for compression ignition as load increases.

Conventional fuels allow a limited scope for improvement of the combustion processes in petrol and diesel engines. In both cases, the ability to extend the operating range of homogeneous compression ignition is heavily dependent on fuel properties. Research into a new fuel specification that offers extended capabilities therefore represents a logical step. The next question is whether this should be developed on the basis of a petrol or diesel fuel with the corresponding engine hardware. A key factor in this issue are the requirements the fuel must fulfil at the higher and lower mean pressures, which are more similar to the conditions of a diesel engine. For instance, an increase in the flammability of petrol in the upper load range would inevitably lead to engine knock. In a diesel engine, however, a lowering of the boiling ranges is advantageous even at full load. In addition, compression ignition at all loads in a diesel engine ensures maximum combustion efficiency when the engine leaves the homogenisation range. The development objective is thus not just a reduction in emissions, but efficient combustion with minimum pollutant emissions. The fuel matrix being studied within research into CCS is shown in the **Table**. Compared to the reference fuel (CEC Diesel), the CCS Kerosine used so far has a lower cetane number and also better vaporisation properties. Both changes lead to an improved homogenisation, and thus lower emissions and higher efficiency. A next optimisation step is now to investigate the fuel CCS Naphtha. Compared with CCS Kerosine, its vaporisation characteristics are even better.

2 Fuel Variation with a Single-cylinder Engine

For investigation of and research into the CCS process and possible synthetic fuels a

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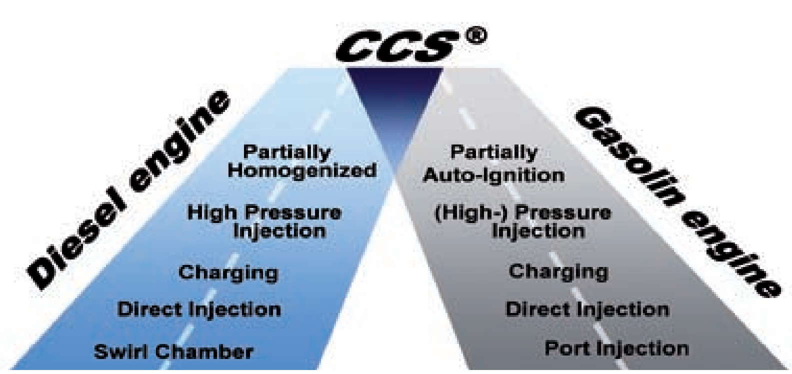


Figure 1: From diesel and petrol combustion to the CCS process

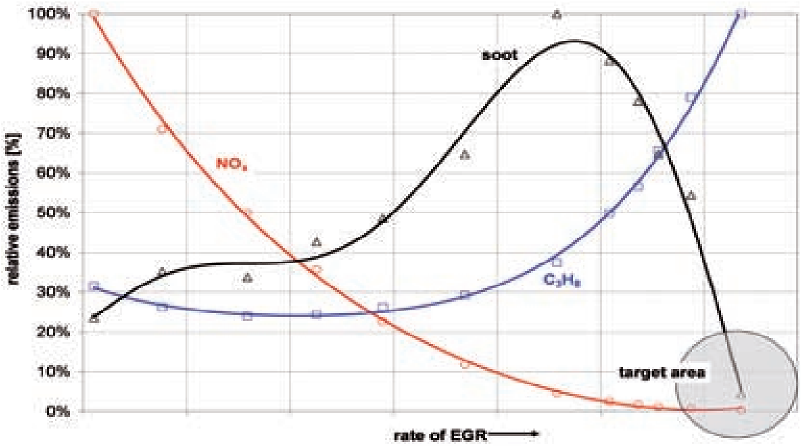


Figure 2: Effects of homogenisation on the formation of pollutants in diesel engine combustion (engine speed 1600 rpm; injection quantity 12 mg/stroke; pmi 3.3 bar)

defined process chain is used [3]. With this it is possible to analyse and optimise the complex system of interactions consisting of injection, ignition, premixed and diffusion combustion, as well as pollutant formation. Process simulations are carried out in parallel to support the investigations [4]. The process chain for combustion process development begins with component tests, which examine aspects such as mechanical properties. Injection, spray break-up and evaporation can be analysed by means of visualitation in a diesel testing pressure chamber, which reproduces the thermodynamic conditions of the engine cycle. Combustion is also examined with optical access in a transparent engine, which allows assessment of emission formation. Single-cylinder engines are used for component examinations under realistic engine conditions. The process chain is completed by investigations in dynamic full engine test beds and finally by vehicle prototypes to examine dynamics and emissions characteristics in driving cycles.

A fully conditionable single-cylinder engine is used for testing the influence of the fuel on engine combustion. The single-cylinder research engine is equipped with an externally powered common rail injection system, supplying injection pressures of up to 1800 bar, and has two intake valves and one exhaust valve. It has a cylinder displacement of 492 cm^3 and a compression ratio of 16.5. To ensure reproducible results, it has conditioning systems for the engine oil and coolant temperature, and the intake pipe pressure and temperature. A valve in the exhaust pipe is used to adjust the backpressure. Various fuel specifications from the Table are tested using the single-cylinder research engine. The investigations are focused on extending the operating range of homogeneous compression ignition. The particulate matter emissions are used as the measure of homogenisation.

Figure 3 shows the particulate matter emissions for the fuels CEC Diesel, CCS Kerosine and CCS Naphtha over a varying air fuel ratio for a constant mean indicated pressure of p_{mi} = 6.9 bar at 2000 rpm. The intake pipe pressure, intake pipe temperature, exhaust backpressure and rail pressure are constant. The air fuel ratio is adjusted by raising the exhaust gas recirculation rate. Starting from high air fuel ratios, the particulate matter emissions start to rise at λ = 1.8 for the three fuels shown here. The highest emissions were measured in operation with CEC Diesel. Only at λ = 1.05 is a slight fall in the particulate matter level recorded, but from a high level. When using CCS Kerosine, the reduction in the air fuel ratio results in an appreciably lower rise in particulate matter. Moreover, particulates already begin to decrease again at λ = 1.2. When the engine is run on CCS Naphtha, no notable increase in particulate matter emissions is found as the air fuel ratio is varied. At a mean indicated pressure of p_{mi} = 14.4 bar and 2000 rpm, the particulate matter emissions at varying air fuel ratios for the three fuels tested are shown in **Figure 4**. As the air fuel ratio is lowered, particulate matter emissions rise continuously for the fuels shown; they were not found to subsequently decrease again. However, the relative trend between the fuels remains the same: lower particulate matter emissions are achieved with CCS Naphtha than with CCS Kerosine or with CEC Diesel.

For all results in Figure 3 and Figure 4, the rise in particulate matter emissions when the air fuel ratio is reduced can be explained by the lower partial pressure of oxygen; due to insufficient homogenisation, local regions with insufficient oxygen occur. Nevertheless, the synthetic fuels show a significant advantage over CEC Diesel. At low loads, as shown in Figure 3, and short necessary injection durations, the recirculation of exhaust gas can considerably increase the ignition delay. Due to long ignition delays the improved homogenisation has the effect of overcompensating for the lack of oxygen and the particulate matter emissions decrease in spite of the further reduction in the air fuel ratio. The fuels‘ homogenisation potential can be extended through the described measures of lowering the cetane number and the boiling point, **Table**.

The minimum achievable NO_x emissions in the series of measurements dis-

Table: Fuel properties

	CEC diesel	CCS Kerosine	CCS Naphtha
Cetane number CFR [-]	56	45	45
Initial boiling point 5 % [°C]	200	160	77
Final boiling point 95 % [°C]	350	210	161
Density [kg/m³]	835	778	689
Net calorific value [MJ/kg]	42.8	43.5	44.6
Aromatics [% weight]	20.4	< 1	< 1

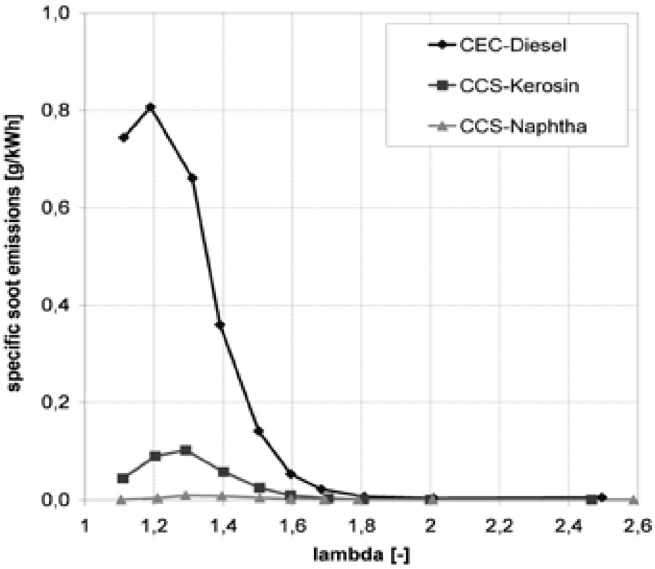


Figure 3: Partially homogeneous point PM – NO_x trade-off (engine speed 2000 rpm; injection quantity 16.9 mg/stroke; p_{mi} 6.9 bar)

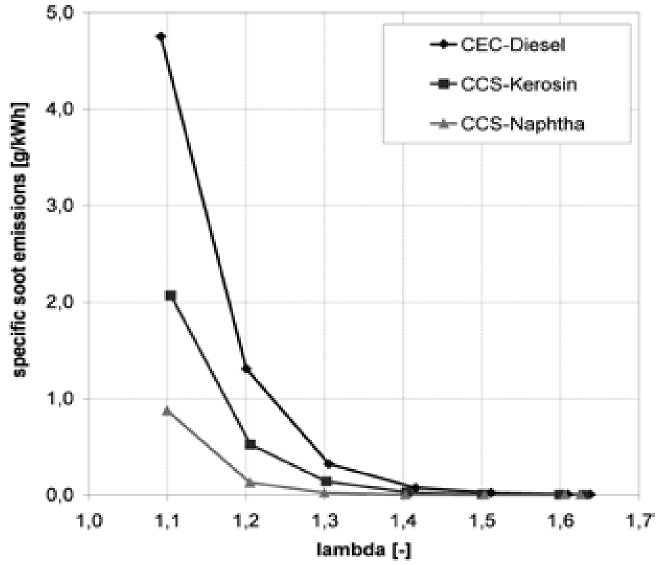


Figure 4: Conventional point PM – NO_x trade-off (engine speed 2500 rpm; injection quantity 35.1 mg/stroke; p_{mi} 14.4 bar)

cussed show no differences between the fuels. As known, by increasing the quantity of recirculated exhaust gas, the NO_x emissions can be continuously reduced.

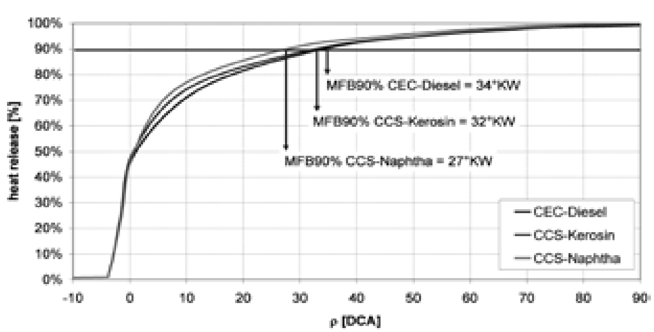
The homogenisation has an adverse effect on HC emissions. As the boiling point is reduced, there is an increase in the scale of lean zones in the combustion chamber, which cannot be burnt due to, for instance, proximity to the combustion chamber wall. In a cold engine, hydrocarbon emissions are 15 % higher with the CCS Kerosine and 30 % higher with CCS Naphtha than with CEC Diesel. However, this increase can be mitigated by means of a future optimisation of the combustion chamber geometry, the charge motion and the injector in combination with a modified oxidation catalyst.

The use of synthetic fuels can substantially reduce CO_2 emissions. Their advantages stem from the thermodynamically favourably position of the 50-% energy conversion position, shortly after TDC. In **Figure 5** the combustion curves for CEC Diesel, CCS Kerosine and CCS Naphtha for the load point p_{mi} = 6.9 bar at 2000 rpm and λ = 1.4 demonstrate the efficient position of combustion achieved. After a uniform start of combustion at approx. 4°KW after TDC, the combustion rates for CEC Diesel, CCS Kerosine and CCS Naphtha are initially identical. At roughly 10°KW after TDC, the combustion curves separate with CEC Diesel burning more slowly than CCS Kerosine and CCS Naphtha. As a result, the point at which 90 % of fuel energy has been converted (AI90%) is reached earlier in the cases of CCS Naphtha and CCS Kerosine than with CEC Diesel. The faster energy conversion leads to an increase in thermal efficiency. This consequently reduces the required injection quantity, resulting in lower CO_2 emissions. With the 50-% energy conversion position at the thermodynamic optimum, CO_2 emissions can be additionally reduced by 4 % using CCS Kerosine and 8 % using CCS Naptha compared with CEC Diesel. This includes a reduction in CO_2 emissions resulting from the H/C ratio (CEC diesel: 1.85; CCS Kerosine: 2.05; CCS Naphtha: 2.29).

An emissions number was defined for the evaluation of the results of the stationary fuel tests on the single-cylinder engine. This emissions number takes into account particulate matter, NO_x and HC, these being emissions tested in the driving cycle, as well as fuel consumption. It is based on the ratios between the emissions in the current and future exhaust emissions legislation.

Figure 6 shows the emissions numbers on a logarithmic scale of 10 stationary operating points for the fuels CEC Diesel, CCS Kerosine and CCS Naphtha. The lower the emissions number, the lower the emissions (particulate matter, NO_x, HC) and fuel consumption. With the exception of the operating points 40 Nm at 1200 rpm and 60 Nm at 2500 rpm, CEC diesel is found to yield the highest emissions number, followed by CCS Kerosine and CCS Naphtha. At the operating point 40 Nm at 1200 rpm, the lowest emissions number is achieved with CCS Kerosine; the same applies to the operating point 60 Nm at 2500 rpm. Over wide sections of the engine map, the use of CSS-Naptha can reduce the emission of

Figure 5: Analysis of the combustion processes with CEC Diesel, CCS Kerosine and CCS Naphtha



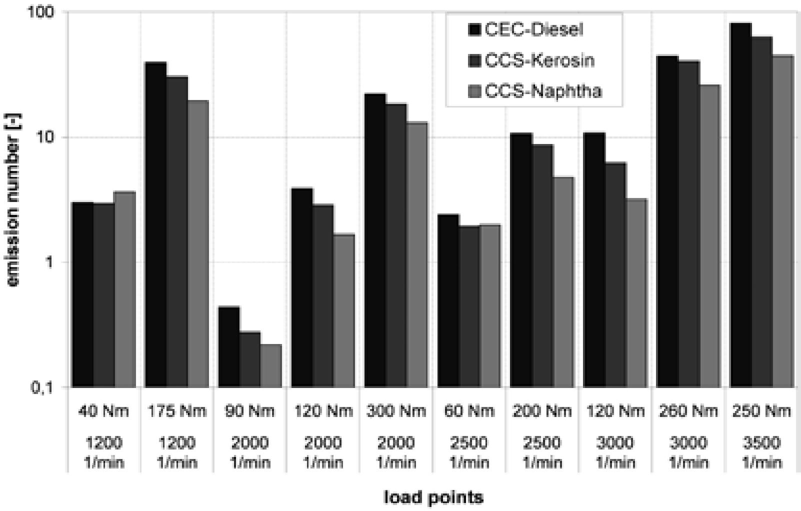


Figure 6: Assessment of potential based on the emissions number

substances that are subject to emissions limits and improve fuel economy as the result of improved fuel properties with regard to evaporation characteristics and cetane number.

3 Description of the CCS Combustion System in a Four-cylinder Engine

The fuel CCS Naphtha has not yet been used in a full engine. The following section describes the combustion system with CCS Kerosine, including functionalities, regulation strategies and, in particular, the early-homogeneous operating mode. In conventional diesel combustion systems, at least one pilot injection is used for acoustic reasons. Emissions of substances subject to limits are reduced by means of recirculation of moderate quantities of exhaust gas and increasingly shifting the 50-% energy conversion position positions further after TDC. To some extent, the effects of extending ignition delay can be used to achieve homogenisation and NO_x reduction. This, however, has a significant detrimental effect on efficiency. The early-homogeneous combustion process is characterised by a single injection well in advance of TDC. Very high, controlled rates of exhaust gas recirculation are used to achieve a thermodynamically favourable position of combustion, shortly after TDC. In addition, the exhaust gas recirculation lowers the reaction speed, thus reducing combustion noise. The advantage of the early/homogeneous combustion process in terms of emissions stems from the long ignition delay time, which leads to an improved mixture formation.

As described in chapter 1, as load increases, a limit is reached, above which the early-homogeneous combustion process can no longer be maintained due to the increasing injection quantity. An operating mode change-over functionality must therefore be provided, which results in no change in torque for the driver. As an example of the operating mode change-over in the CCS process, **Figure 7** shows injection strategy on a cross-sectional graph of engine loads at 2000 rpm. It shows the start of activation for the main injection (round symbols) and pilot injection (rectangular symbol) as a function of the effective torque. The early-homogeneous operating mode (gray) with just one main injection, as opposed to the conventional operating mode (black) with a pilot and a main injection, can be implemented in the range between 0 Nm and approx. 100 Nm.

The basic difference in the quantity of recirculated exhaust gas required for the conventional and early-homogeneous operating modes represents a problem for the engine application. In addition, different injection strategies are used. The operating mode change-over from conventional to early-homogeneous requires a sufficient quantity of recirculated exhaust gas to be present in the intake manifold. In order to deliver the very high exhaust gas recirculation rates of up to 70 %, the high-pressure EGR fitted in production engines is supplemented by an additional low-pressure EGR. When the required EGR rate is available, the change-over from the conventional to the early/homogeneous operating mode can take place. As this happens, the 50-% energy conversion position is regulated in real time to a position shortly after TDC for optimum efficiency. The possible efficiency gains of the early-homogeneous operating mode are shown in **Figure 8**, mapping the fuel consumption difference between conventional and early-homogeneous operating modes. In the engine speed range of 1000 rpm to 3000 rpm and at up to 100 Nm, the engine control system changes over to the early/homogeneous operating mode when the required EGR quantity has

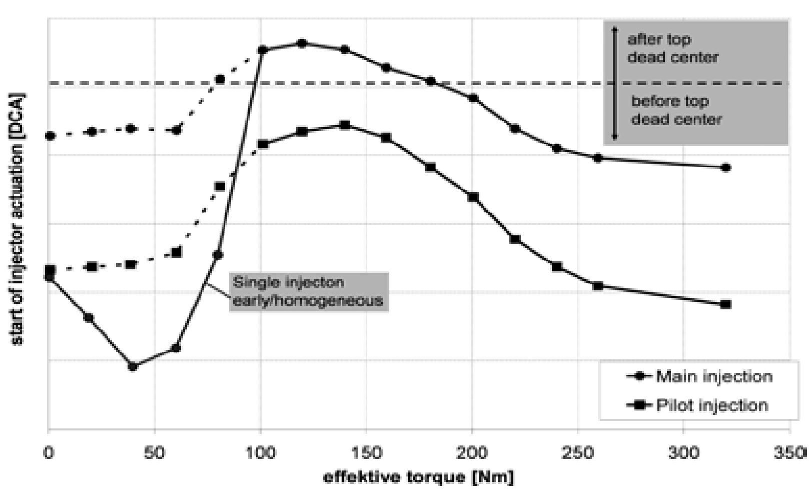


Figure 7: Injection strategy shown over an engine load cross section

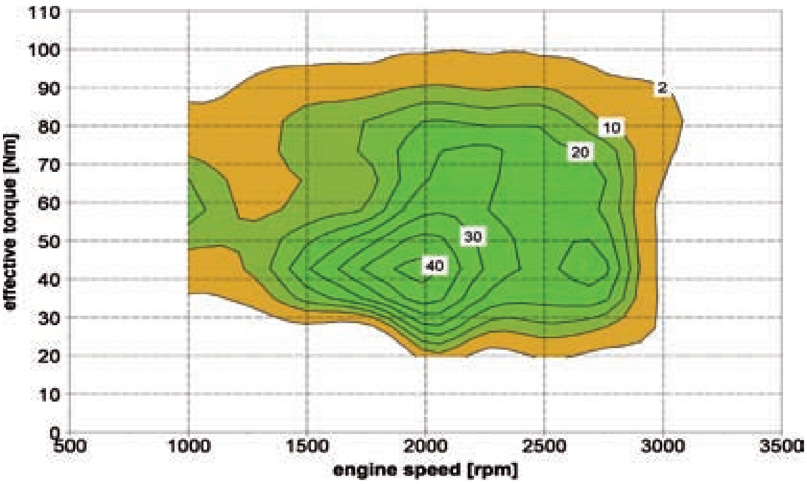


Figure 8: Fuel consumption advantage [g/kWh] of the early-homogeneous operating mode compared to the conventional operating mode

been reached. Above this load, the engine is currently operated with the conventional combustion process.

The increase in efficiency in the early-homogeneous mode results from the optimum-efficiency position of combustion. As a consequence, the application of the early-homogeneous operating mode in the VW Touran prototype increases efficiency in the New European Driving Cycle NEDC by up to 8 %. In real operation, driving the vehicle at up to approx. 100 km/h, efficiency was increased by up to 40 g/kWh. This leads to a significant reduction in CO_2 emissions.

With future use of the fuel CCS Naphtha, it is possible to further improve the CO_2 advantage, because the end of combustion is reached earlier with CCS Naphtha while the 50-% energy conversion position remains the same, Figure 5. Moreover, its improved homogenisation properties allow an extension of the load range in which the early-homogeneous operating mode can be implemented.

Alongside increased efficiency, the implementation of the CCS Combustion System has the potential to simultaneously lower particulate matter and NO_x emissions, Figure 2. It thus offers a means of fulfilling future emissions limits.

Figure 9 shows a comparison of the CCS prototype's normalised NO_x emissions for the NEDC using the conventional diesel combustion systems only (green) based on EU IV and the CCS Combustion System with use of the early/homogeneous operating range (red). The speed in the NEDC is also shown. The CCS Combustion System yields a reduction in NO_x emissions of approx. 65 %. Particulate matter emissions upstream of the diesel particulate filter are also halved in comparison to the production application.

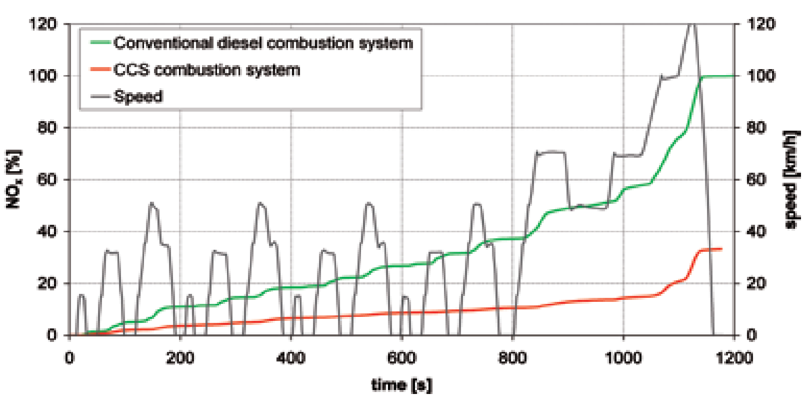


Figure 9: NEDC result for the CCS prototype

4 Summary and Outlook

Alongside their fulfilment of future exhaust gas legislation, new internal combustion engines will make a major contribution to reducing CO_2 emissions. Concepts for the further development of diesel and petrol engines with conventional fuels are being intensively investigated by the Group Research - Drive System of VW, extending to their implementation in concept vehicles [1], [2], [4]. Intensive research is being carried out into new fuel specifications with the objective of extending the early/homogeneous operating range. The synergetic use of the advantages of the diesel and petrol engines in combination with modified fuels is referred to as the CCS Combustion System. It specifically combines the homogenisation of fuel and air, a feature of the petrol engine, with the efficient compression ignition of a diesel engine. A research prototype based on a VW Touran was developed for a dynamic implementation of the CCS. In the NEDC, it achieved a reduction in CO_2 emissions of 8 % compared with the Euro 4 production application while NO_x was lowered by 65 %. Fuel testing, aiming to extend the early homogeneous operating range, is carried out on the single-cylinder research engine test rig. CCS Kerosine and CCS Naphtha also have the potential to further lower particulate matter and NO_x emissions. Alongside the efficiency gains of the CCS, the fuel's favourable H/C ratio provides a further CO_2 advantage.

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