Adaptive Control for the Real Driving Emissions of Diesel Engines

The operating strategy for minimising pollutant formation is usually frozen as soon as a diesel engine leaves production. This means that the raw emission characteristic of the engine cannot be changed anymore. At the ETH Zurich, the benefits of a variable operating strategy for the internal motor actuators of a diesel engine were investigated within the framework of an FVV project, which could be used to adapt the raw emissions behaviour in a situation-specific manner.

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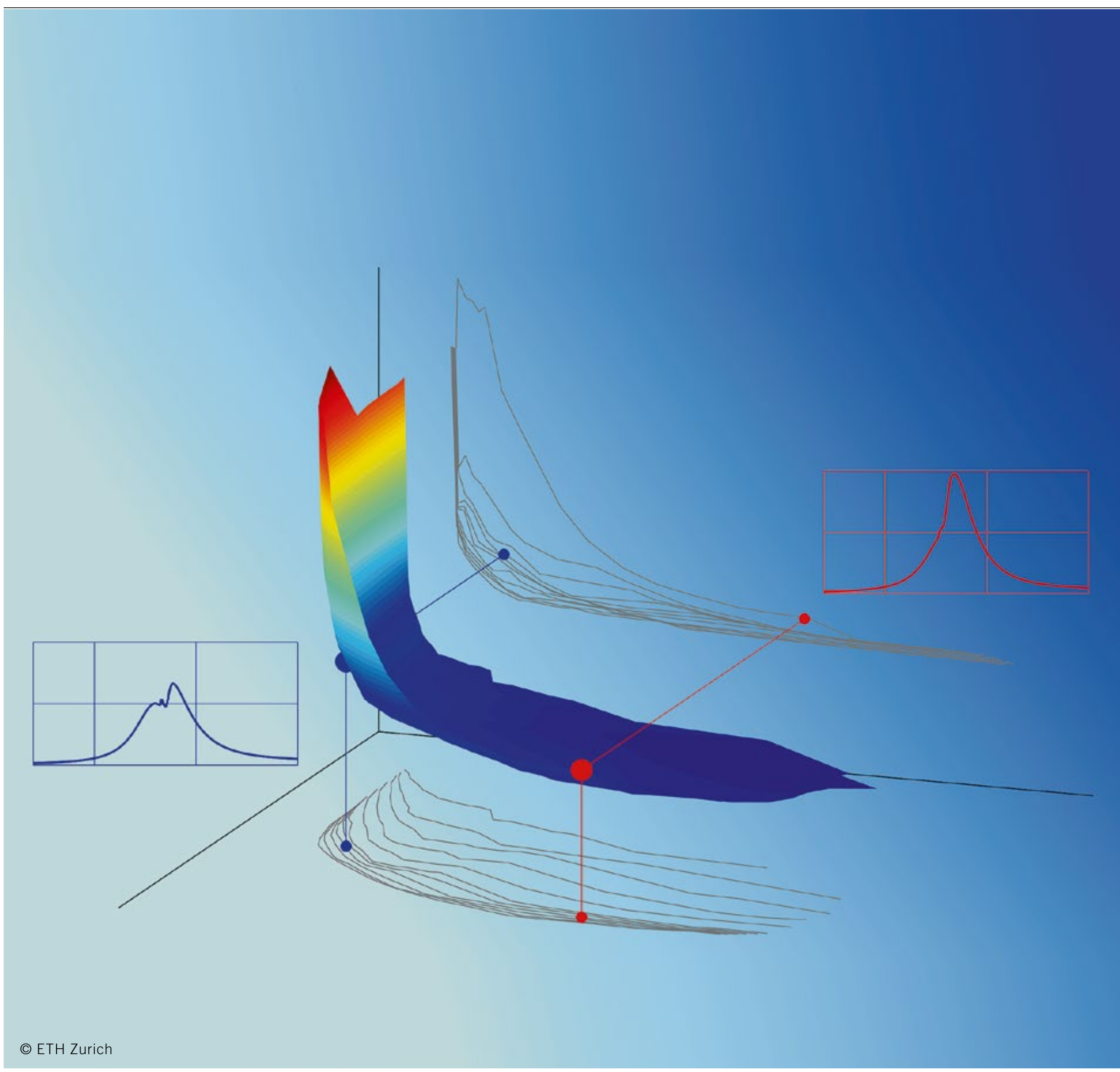


1 MOTIVATION

The ever-increasing demand on the purity of vehicle exhaust gases [1] has sparked the further development of diesel engine systems among manufacturers. Most passenger car diesel engines use in-engine measures to minimise pollutant formation, i.e. exhaust gas recirculation and injection timing [2], as well as a chain of exhaust gas aftertreatment systems to clean tailpipe exhaust gases [3]. The minimisation of pollutant formation requires the careful tuning of the in-engine actuators, which is a delicate task that is difficult to automate [4]. Furthermore, manufacturers freeze this in-engine actuation strategy, once the engine leaves production, which means that the raw emission characteristic of the engine cannot be changed anymore.

2 HIERARCHICAL STRUCTURE OF THE CONTROL SYSTEM

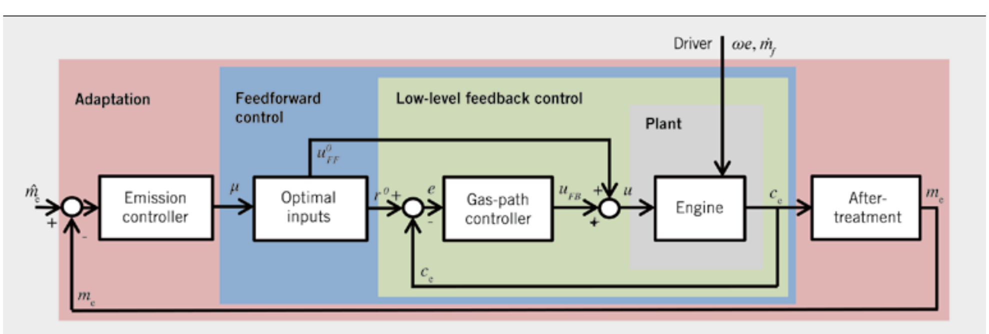
In the following, the opportunities offered by a variable in-engine actuation strategy are investigated. Such a strategy could be used to influence the raw emission characteristic of the engine during operation. In a so-called real driving emission (RDE) test [5, 6], the engine emission characteristic could be adjusted during operation according to the needs of the exhaust gas aftertreatment systems. Of course, legislation may have to be adjusted in order to allow for such an integrated emission management [7, 8], nevertheless, this article simply aims at investigating its potential benefits in terms of fuel consumption reduction and minimisation of tailpipe emissions. The systematic procedure was developed within the scientific research project Emission Optimised Diesel Engine sponsored by the FVV [9]. The methodology starts by introducing a clear hierarchical structure into the engine control system, **FIGURE 1**.

The innermost and fast low-level feedback control loop is responsible to ensure the best possible reference tracking performance for the boost-pressure, the exhaust-gas recirculation rate, etc. Rather than to find individual optima for each feedback control loop, the task of this hierarchy level is to minimise the control errors. Correspondingly, classical design methods can be used to synthesise these controllers. The slower, intermediate hierarchy level termed feedforward control is responsible for the optimal coordination of all low-level control loops, by issuing both optimal reference and feed-forward control signals. The aim is, to implement the family of all optimal in-engine actuation strategies, which in turn allows to adapt the raw emission characteristic of the engine at any given point in time. The even slower, outermost adaptation loop makes use of the new degree of freedom to adjust the engine raw-emission characteristic according to the current driving style or the needs of the aftertreatment systems. A number of ways to implement such an adaptation controller are conceivable. This article discusses an RDE controller for a diesel engine passenger car.

It is suggested, to derive the optimal feed-forward controller based on a numerical optimisation using the results from a steady-state measurement campaign. Here, a full-factorial variation of all relevant actuators (that is to say, including all cross variations) has been used to gather the data required for the optimisation. Of course, methods from the design-of-experiment theory [10, 11] could be used in order to reduce the necessary test-bench time. The experimental results from an engine-in-the-loop emulation of the New European Driving Cycle validate that close to optimal results can be achieved, despite the fact that the engine dynamic behaviour has been neglected in the derivation of the controller, and that an online adaptation of the engine emission characteristic during operation is possible at any given point in time. The proposed procedure is systematic and repeatable, integrates well with existing controllers and measurement practice, and requires only little background knowledge in numerical optimisation.

The experiments in this study have been conducted using a highly dynamic engine test-bed. The Daimler OM642 six-cylinder, 3-l V engine that first went into series production in 2005 was operated without any exhaust gas aftertreatment devices. A rapid prototyping module of the type ES910 by Etas was used to send external control signals to the engine actuators. To analyse the emission characteristics of the engine, exhaust gas sensors for soot and nitrogen oxides have been used. In this study only the fully warmed-up engine was considered and only the most relevant actuators have been taken into account. In principle, however, the method could be extended to cover additional actuators, and to be applicable in cold-start conditions as well.

FIGURE 1 Hierarchical structure of the engine control system (© ETH Zurich)



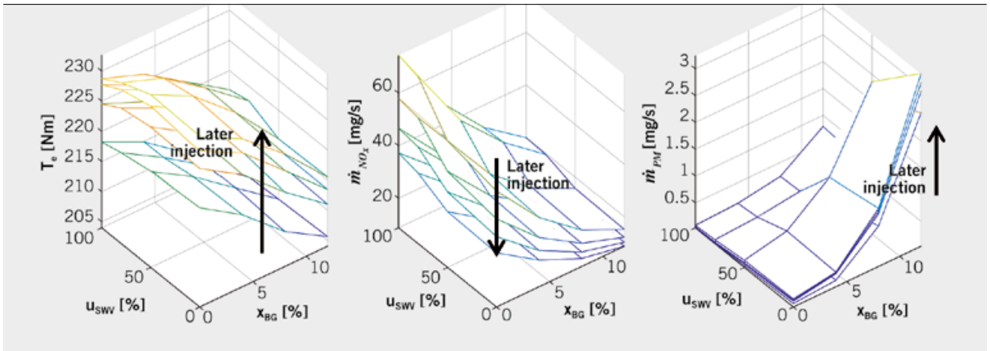


FIGURE 2 Engine response in terms of torque generation (left), nitrogen-oxide (middle) and soot emissions (right) at 1250 rpm and 40 % load (© ETH Zurich)

3 STEADY-STATE PERFORMANCE

The steady-state performance of the engine is evaluated using a grid measurement campaign, where 31 different operating points are analysed by applying full-factorial cross-variations of all relevant actuators. Only the most relevant actuators are used in this study, that is to say the burnt gas ratio x_{BG}, the swirl valve closing angle u_{SWV} and the start of injection u_{SOI}. Four discrete values are chosen for each actuator, resulting in 64 measurement points per operating point and a total of 1984 measurement points. To conduct the cross-variations, the desired engine speed n is passed to the test-bench speed controller, while the total amount of fuel to be injected $\dot{m}_f$ is passed to the engine electronic control unit. A purposeful choice of nesting the cross-variations is necessary to achieve short measurement sequences. The variation of the burnt-gas ratio is placed in the outer loop (slowest response time) while the variation of the start-of-injection is placed in the inner loop (fastest response time). With this choice, the cross-variations within one working point (n, $\dot{m}_f$) can be executed within 12 min. In total, the steady-state grid measurement campaign can be finished within 8 h. All recorded data are functions of the operating point and the actuators, e.g., exhaust mass flows of the nitrogen oxide $\dot{m}_{NOx}$ and soot emissions $\dot{m}_{PM}$ and the resulting engine torque T_e.

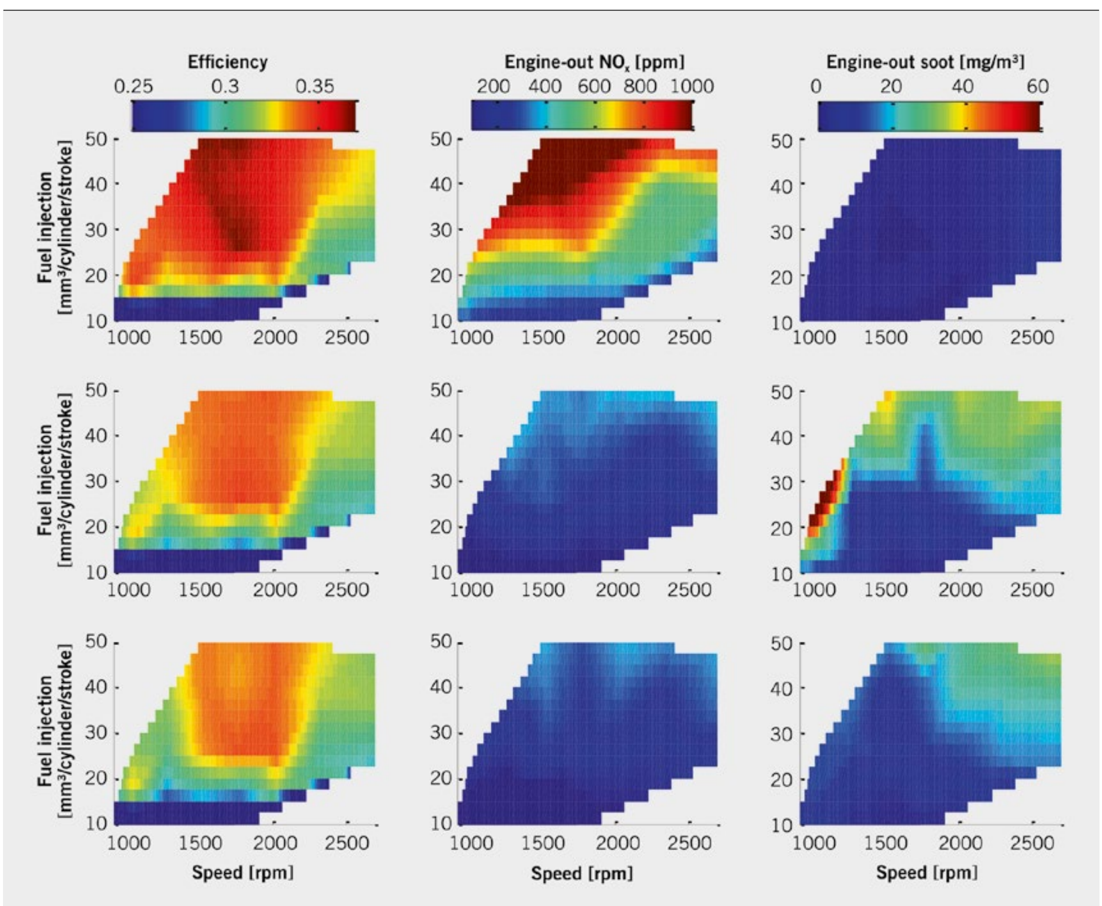


FIGURE 3 Efficiency, nitrogen oxide and soot emissions as a function of the operating point and the strategic value (top: fuel optimal, middle: low NO_x without soot penalty, bottom: low NO_x with soot penalty) (© ETH Zurich)

$$\dot{m}_{NO_x} = f_{NO_x}(n, \dot{m}_f, x_{BG}, u_{SOI}, u_{SWV}) \quad \text{(Eq. 1)}$$

$$\dot{m}_{PM} = f_{PM}(n, \dot{m}_f, x_{BG}, u_{SOI}, u_{SWV}) \quad \text{(Eq. 2)}$$

$$T_e = f_T(n, \dot{m}_f, x_{BG}, u_{SOI}, u_{SWV}) \quad \text{(Eq. 3)}$$

Depicting this data as in **FIGURE 2** reveals the following expected trends: Increasing the burnt-gas ratio yields lower nitrogen oxide emissions, however at the cost of a decreased torque output and increased soot formation. Delaying the fuel injection has a similar effect, however soot formation is barely influenced by the injection timing. When closing the swirl valve, the torque generation and the nitrogen oxide formation experience a minor influence only, a reduction of the soot formation is prominent at higher values of the burnt-gas ratio.

4 OPTIMISATION OF IN-ENGINE CONTROLS

In order to calculate the family of all optimal raw-emission strategies based on the steady-state measurements, an optimal control problem [12] has to be formulated as follows. The goal is to minimise the cumulative fuel consumption over a driving cycle of with a duration of t_f, Eq. 4. Since the engine is approximated as a static system, the accumulation of the nitrogen oxide and soot emissions, Eq. 5, are the only two dynamic state variables. The legislative emission limits $\hat{m}_{NOx}$ and $\hat{m}_{PM}$ prescribe a certain not-to-exceed amount of nitrogen oxides and soot for a given driving cycle, which results in a limitation of the final value $x(t_f)$, Eq. 6. As the choice of control inputs has an influence on the resulting torque generation, an additional constraint is required to ensure that the torque prescribed by the driving cycle $T_{cyc}(t)$ is always fulfilled, Eq. 7.

$$\min_{u(\cdot)} \int_0^{t_f} \dot{m}_f(t)\,dt \quad \text{(Eq. 4)}$$

$$\dot{x}(t) = \begin{bmatrix} f_{NO_x}(n, \dot{m}_f, x_{BG}, u_{SOI}, u_{SWV}) \\ f_{PM}(n, \dot{m}_f, x_{BG}, u_{SOI}, u_{SWV}) \end{bmatrix} \quad \text{(Eq. 5)}$$

$$x(t_f) \leq \begin{bmatrix} \hat{m}_{NO_x} \\ \hat{m}_{PM} \end{bmatrix} = \hat{m} \quad \text{(Eq. 6)}$$

$$T_e(t) = f_T(n, \dot{m}_f, x_{BG}, u_{SOI}, u_{SWV}) = T_{cyc}(t) \quad \text{(Eq. 7)}$$

Applying Pontryagin's Minimum Principle to this optimal control problem yields the dual problem of minimising the Hamiltonian function, Eq. 8. The torque constraint, Eq. 7, can be handled implicitly, by normalising all mass-flows and expressing them in g/kWh. Here, the Hamiltonian function is reformulated such that the dimensionless Lagrange multipliers $\mu_{\#}$ come to lie within the interval [0, 1]. The multiplier μ_{PM} determines the trade-off between soot emissions and fuel consumption, while the second multiplier μ_{NOx} determines the trade-off between nitrogen oxide emissions and the first trade-off. The Lagrange multipliers can be chosen freely and thus reflect the user's preference. In the following, the choice of the multipliers will be termed as emission strategy or simply as strategy. The optimal control inputs

$$u^o = [x^o_{BG}, u^o_{SOI}, u^o_{SWV}]^T \quad \text{(Eq. 8)}$$

are evaluated by a minimisation of the Hamiltonian function and then stored in four-dimensional look-up tables as functions of the operating point and of the emission strategy, Eq. 10.

$$H = (1 - \mu_{NO_x})\left[(1 - \mu_{PM})\,\dot{m}_f + \mu_{PM}\,\dot{m}_{PM}\right] + \mu_{NO_x}\,\dot{m}_{NO_x} \quad \text{(Eq. 9)}$$

$$u^o(n, \dot{m}_f, \mu_{NO_x}, \mu_{PM}) = \operatorname{argmin}_{u(.)}\left\{H(n, \dot{m}_f, u, \mu_{NO_x}, \mu_{PM})\right\} \quad \text{(Eq. 10)}$$

FIGURE 3 shows how the raw emission characteristics of the engine can now be influenced via the optimised control inputs, Eq. 10. The expected performance of the engine is calculated by combining the steady-state measurements from step one with the optimised control inputs. **FIGURE 3** shows the calculated efficiency (left column) as well as the concentrations of nitrogen oxides (centre column) and soot (right column) in the exhaust gas as a function of the engine speed and the fuel injection when using three different optimal raw-emission strategies: fuel optimal (top row), low-NO_x without soot penalty (centre row), and low-NO_x with soot penalty (bottom row). The first strategy leads to good efficiency and low soot emissions, while nitrogen oxide emissions tend to be very high. The second strategy sacrifices some of the fuel efficiency for lower nitrogen oxide emissions, but has a tendency to cause strong soot formation, especially in the low-speed/high-load regime. The last strategy represents a trade-off, where all criteria come to lie in a reasonable range.

5 INTEGRATION INTO THE ENGINE CONTROL SYSTEM

The four-dimensional look-up tables, Eq. 10, containing the optimal control inputs can be directly integrated into the engine control system. Based on the current values of engine speed and injection quantity, as well as the chosen emission strategy parameters μ, interpolation yields the optimal actuation values for μ, as well as the corresponding desired values of the exhaust gas concentrations of nitrogen oxides and soot

$$c_e = [c_{NO_x}, c_{PM}]^T \quad \text{(Eq. 11)}$$

(feedforward control in **FIGURE 1**). In order to validate that the omission of the engine dynamics in the derivation of the optimal emission strategies does not introduce major errors, an experimental test is conducted. A vehicle emulation is used to calculate the speed and torque trajectories that the engine would have to deliver if a 1500 kg vehicle with an automated transmission was following the transient New European Driving Cycle (NEDC). These trajectories are then followed by the real engine on the test-bench six times, using different values of the penalty factor μ_{NOx} each time, while the penalty factor $\mu_{PM} = 0.3$ is kept constant. In each run, the injection quantity is adjusted based on a feedback controller in order to cancel out the distortion of the torque generation introduced by varying the emission strategy.

FIGURE 4 shows the measured cumulative values of the mechanical energy delivered at the crankshaft (top left graph), the increase of the carbon dioxide emissions with respect to the most fuel-efficient strategy (top right graph), the nitrogen oxide and the soot emissions (bottom row graphs). The amount of mechanical energy showed deviations of less than 0.1 % in all six passes, so the six measurements are comparable. The results illustrate that increasing the value of μ_{NOx} yields a reduction of the engine out nitrogen oxide emissions. As expected, this reduction comes at the price of slightly increased soot emissions, as well as a decrease of the efficiency that leads to a corresponding increase of carbon dioxide emissions compared to the most fuel-efficient strategy. Note that the raw-emissions as shown in **FIGURE 4** are engine-out measurements. In order to achieve compliance with legislative emission limits, the engine has to be combined with appropriately designed aftertreatment systems. The whole engine and aftertreatment system will be considered in section 6.

While this experimental test demonstrates that a steady-state optimisation is useful to influence the engine's raw-emission characteristic, no conclusions about the optimality of the different strategies can be drawn from the graphic representation in **FIGURE 4**. In order to measure optimality, the steady-state measurement data is used along with the optimal control inputs to establish a simulation along the speed and torque trajectory of the NEDC. This way, the true optimal cumulative values of the emissions can be calculated that are to be expected if the engine would behave exactly as in the steady-state measurements.

FIGURE 5 shows a comparison of the simulated and measured values: The thick green lines represent the simulated optimal trade-off curve achievable with a value of $\mu_{PM} = 0.3$ and $\mu_{NOx} \in [0.1]$ The grey thin lines represent the same trade-off curve for different values of μ_{PM}. The cumulative values gathered from the experimental test are shown as thick circles, together with the corresponding target values (small circles). The results illustrate that optimality is indeed maintained, even if the engine dynamics are neglected in the optimisation of the control strategies. Furthermore, the engine emission characteristic can be precisely tuned according to the requirements that may arise from legislation and the design process of the aftertreatment system. For example, a given target value for the cumulative engine-out nitrogen-oxide emissions can be met with a precision of 5 %.

This very same experimental test has been repeated, and the positive results have been validated once more, using the Worldwide Harmonized Light-duty Vehicle Test Cycle (WLTC) representing highly transient real-world driving conditions. These results have been presented in the final report of the FVV project 1140 [9].

6 ADAPTIVE EMISSION CONTROL

The following section explains how the proposed variable raw emissions strategy can help minimise the real driving emissions for a diesel passenger car with SCR systems and diesel particulate filters. Based on the steady state measurements and the simulation models of the aftertreatment systems [13, 14], the tailpipe emissions are estimated for the NEDC and two different driving scenarios highway and city that were extracted from the WLTC. **FIGURE 6** shows that the best raw-emission strategy for the NEDC is $\mu_{NOx} = 0.6$ since it is compliant with the Euro 6 limits but does not incur excessive soot and increased carbon dioxide emissions. However, driving on a highway with the same nominal raw emission strategy results in a violation of the nitrogen oxide emission limit. Of course, such a deviation is unacceptable. The obvious solution to this problem, namely to more conservatively design the exhaust

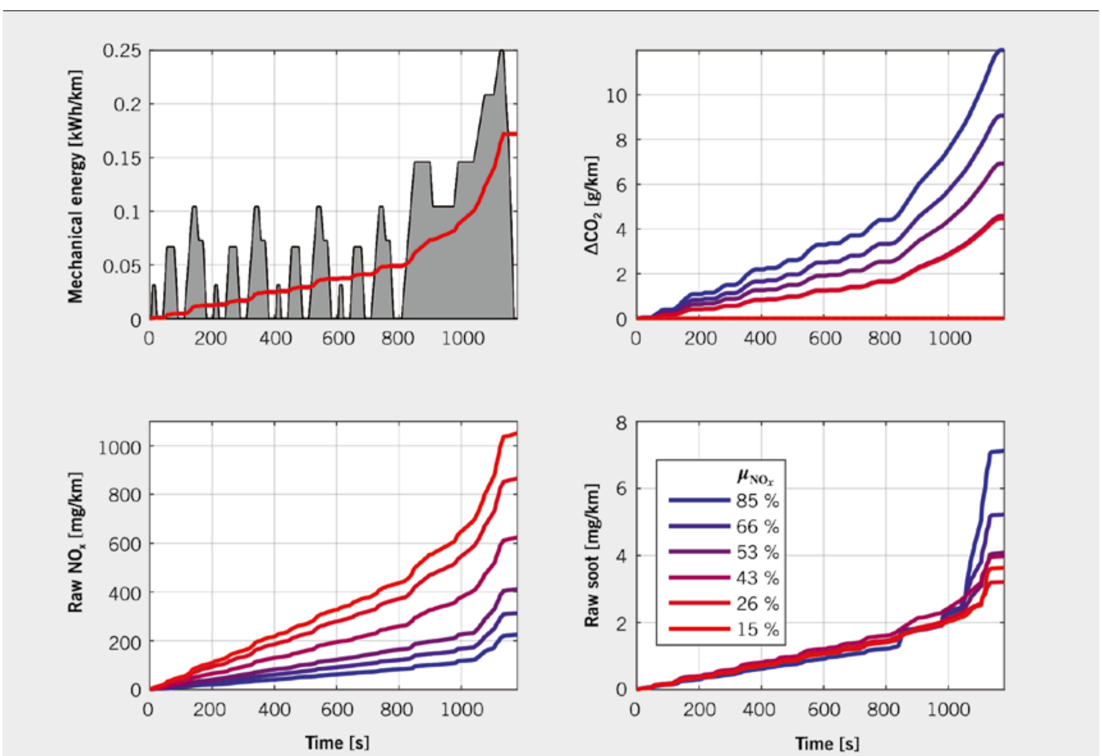


FIGURE 4 Experimental test results of six different raw-emission strategies evaluated on the real engine using the NEDC (© ETH Zurich)

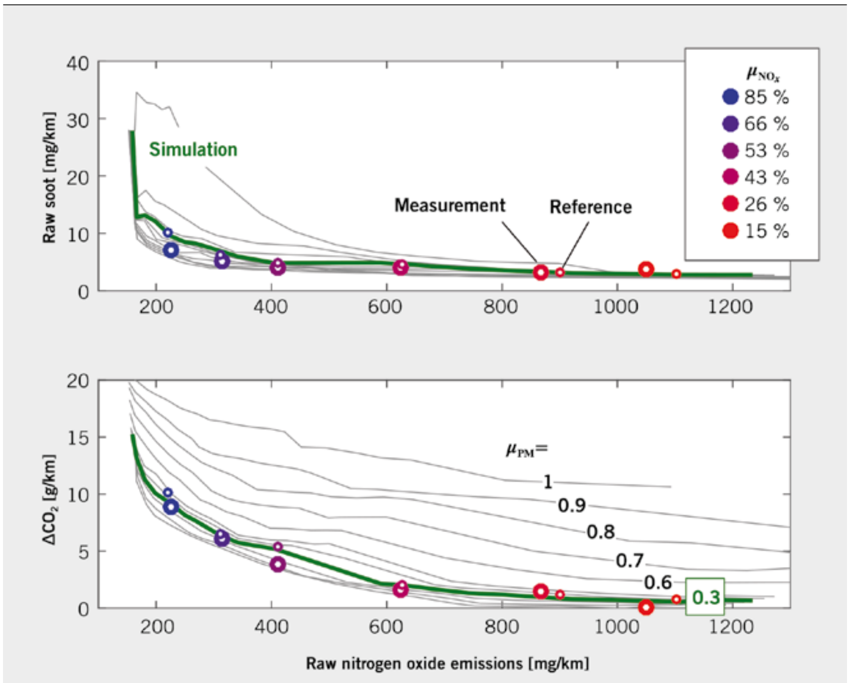


FIGURE 5 Comparison of results from simulation and engine-in-the-loop experiment (the green curve represents the ideally achievable performance of the engine on the NEDC, the thick circles represent the cumulative values measured during the experimental test with six different emission strategies) (© ETH Zurich)

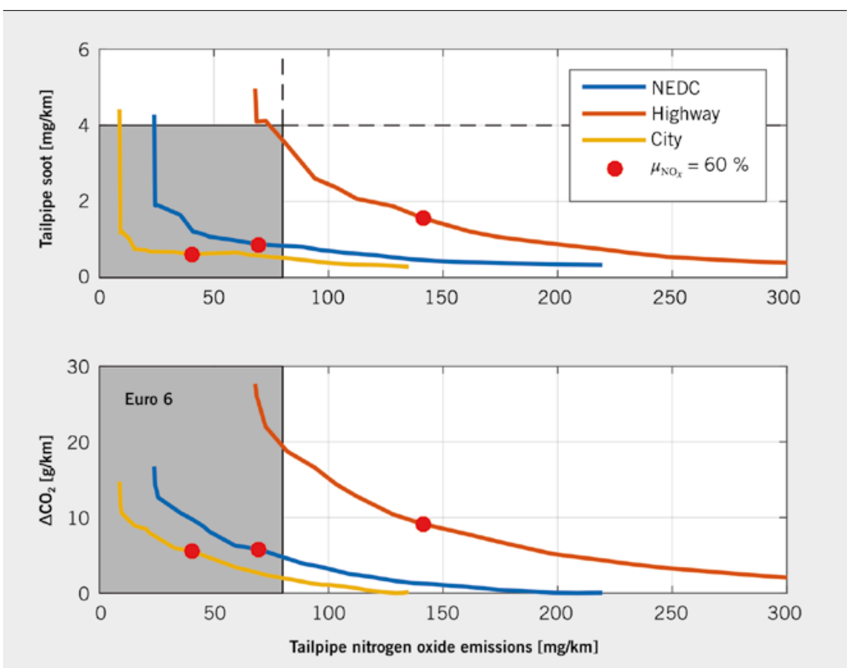


FIGURE 6 Strategies for the NEDC (© ETH Zurich)

aftertreatment systems, is not necessarily the most efficient solution. With the adjustable raw emission control proposed here, the engine's emission strategy can be adjusted and adapted according to the current driving situation. Thus, the procedure offers several possibilities to find the best trade-off between cost and fuel consumption, while complying with the emission legislation.

FIGURE 7 shows the results of a simulation study, where the driving style is changed from city driving to highway driving. Although the nominal strategy (red) complies with the Euro 6 limits in urban traffic, the nitric oxide limit value is exceeded as soon as the vehicle is operated on the motorway. The adaptive strategy (blue), on the other hand, recognises the limit value overshoot and carries out the emission strategy of the engine in such a way that the Euro 6 standard can still be complied with. Compared to the nominal strategy, however, an increase in emissions of soot and carbon dioxide must be tolerated in this case. In the context of the global trade-off between total production cost, fuel economy, CO_2 and consumption of operating fluids, this alternative design path may offer additional optimisation potential, as long as the complete system is designed to comply with the legislative emission limits.

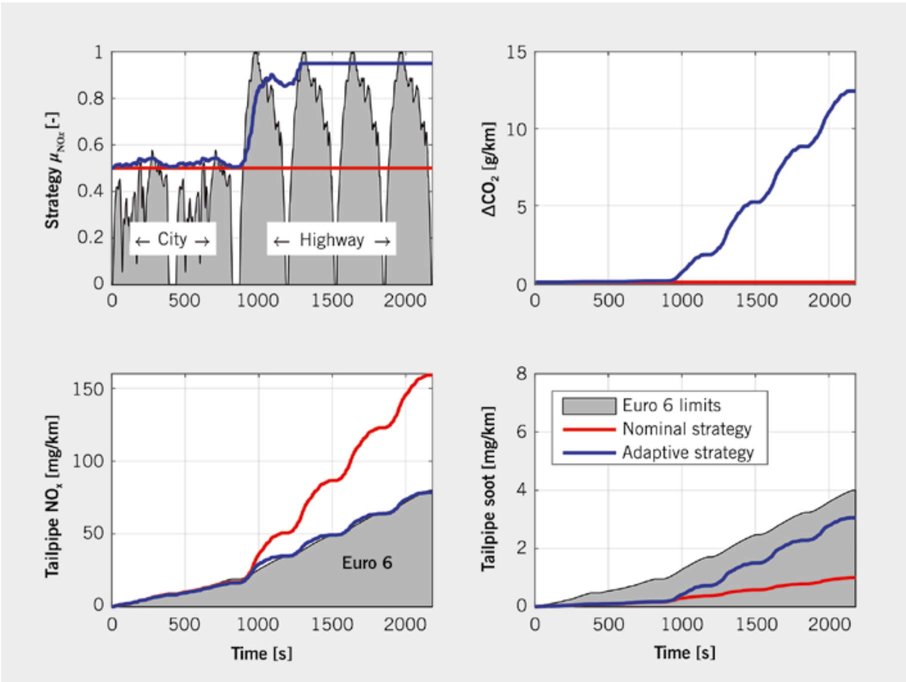


FIGURE 7 Results of a simulation study (© ETH Zurich)

7 SUMMARY AND OUTLOOK

The potential benefits of a variable in-engine actuation strategy were described that could be used to influence the raw emissions a diesel engine during operation. For that purpose, this article describes a straightforward and systematic optimisation method. The method simultaneously evaluates the family of all Pareto-optimal in-engine control strategies. As a result, the engine raw-emission characteristic can be influenced during operation in real-time. The variable raw-emission characteristic of the engine obtained by this optimisation, is demonstrated to be effective in keeping the tailpipe emissions of a complete engine and aftertreatment system below legislative limits in a real driving scenario. Of course, the question, whether such an integrated emission management complies with future emission legislation is still open. The optimisation results have been validated in an experimental test using a Daimler OM642 V6 diesel engine and an engine in-the-loop vehicle emulation.

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