Lowest Engine-Out Emissions as the Key to the Future of the Heavy-Duty Diesel Engine – New Development Results

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ABSTRACT

Latest development results on heavy duty diesel engines show that - applying specific exhaust gas aftertreatment - it is possible to achieve the required U.S. emission limits for 2010. However, based on achieved NOx and particulates engine-out emissions of approximately 1,0 g/kWh NOx and 0,06 g/kWh PM, exhaust gas aftertreatment systems with DeNOx and filter efficiencies of at least 90 % are mandatory. To achieve the mentioned engine-out emissions it is necessary to realize engine concepts with a peak firing pressure capability of at least 200 bar, cooled exhaust gas recirculation with EGR rates up to 25 % at full load, injection systems with a pressure potential of 2500 bar and two-stage turbocharging. Turbocharger efficiencies of more than 50% are required to keep the unavoidable increase of fuel consumption within acceptable limits. For Europe the same applies in case Europe would introduce the U.S. limits. In case the limits for NOx would be set to for instance 1,0 instead of 0,27 g/kWh, the then possible higher engine-out emissions will allow reduced requirements for the engine concept along with a significantly better fuel consumption.

INTRODUCTION

US/EPA 2010 exhaust emission standards for heavy-duty diesel engines are that stringent, that once more the question has to be raised whether such low limits are meaningful or justified at all. Today, development directions for diesel engines as well as for exhaust gas aftertreatment systems depend more than ever on the reasonable setting of emission standards, since in future only relatively small changes of emission limits will have much more crucial impacts on technologies to be applied than it was the case so far. It is the objective of this paper to define and discuss on the basis of newest results of combustion development the main requirements of the basic heavy-duty diesel engine which are needed to meet emission levels of 2010 and beyond. For USA these limits are already defined, whereas for Japan they are being discussed for introduction as early as 2008. In Europe, i.e. for EURO 6, corresponding emission limits do not yet exist. In this paper, the influence of the severity of future exhaust emission limits will be considered on the basis of different technology scenarios and latest development results elaborated at AVL. Their impact on required or possible engine-out emissions as well as on fuel consumption and technology demand will be discussed.

EXHAUST EMISSION LIMITS AND REQUIRED ENGINE-OUT EMISSIONS

Table 1 shows emission limits of the most important markets - for the sake of better comparability expressed in g/kWh. Thus, the US 2010 NOx/PM limits in the Transient Test (Federal Test Procedure) of NOx/PM = 0,2/0,01g/bhp-hr appear as 0,27/0,013 g/kWh. Currently very similar numbers are being discussed in Japan for 2008, i.e. the so-called Japan Post New Long Term Regulation (JPNLTR), requiring NOx/PM values of 0,27/0,01 g/kWh in the new Japanese Transient Cycle (JTC). Since the FTP and even more the JTC exhibit rather low load operating conditions compared to the European Transient Test (ETC), the resulting low exhaust gas temperatures impose special challenges upon the correct functioning of aftertreatment systems in both countries.

USA 2010, USHDTC, ESC
Japan 2008, JTC, under discussion

Component	Emission limit [g/kWh]	Efficiency of aftertreatment system [%]	Max. raw emission [g/kWh]
PM USA	**0.013**	90	**0.1**
PM Japan	**0.01**	90	**0.08**
NOx (USA & J)	**0.27**	80	**1.0**
NOx (USA & J)	**0.27**	90	**2.0**

Europe 2012, ESC, ETC, assumption for EURO6

Component	Emission limit [g/kWh]	Efficiency of aftertreatment system [%]	Max. raw emission [g/kWh]
PM	**0.01**	75	**0.03**
PM	**0.01**	90	**0.08**
NOx	**1.0**	80	**4.0**

Table 1 Future Exhaust Emission Standards and Required Raw Emissions

In general, the following considerations presume the availability of mature exhaust aftertreatment systems in due time. This is certainly the case already today for SCR-systems with the injection of a urea-water solution as NOx-reductant, since with the introduction of EURO 4 some European truck manufacturers are going to bring this technology to the market. But also diesel particulate filters (DPF) as well as NOx-Adsorbers are assumed to become mature for mass production during the remaining time.

Assuming further, that in spite of unfavourable exhaust temperature levels, DPF-efficiencies of 90% and DeNOx-efficiencies of 80% will be achieved, raw exhaust emissions for NOx/PM of 1,0/0.01 g/kWh will be required. As shown in the lower part of Table 1 for Europe, a scenario is taken into account with a NOx limit of 1,0 g/kWh requiring an engine-out NOx level of 4,0 g/kWh. As shown later, the basic engine concept for 4 g/kWh imposes challenges high enough so that it might be interesting to also discuss an engine concept for a NOx limit of 1 g/kWh. At this limit, a DeNOx system with 80% efficiency would be required with a basic engine with raw NOx-emissions of 4 g/kWh. In contrast to the extremely low emission standards in USA, a 1 g/kWh NOx-limit for Europe would allow to consider the use of diverse particulate aftertreatment systems.

In order to arrange further discussions more transparent, the following considerations will be based on a European HD Diesel engine with SCR aftertreatment (w/o DPF) for Euro 4 (3.5 g/kWh NOx) since at this NOx limit the basic engine can be adjusted for best fuel consumption (about 200g/kWh over the ETC test) due to the high efficiency of the SCR-system. However, regarding consumption also the additional cost of the urea-water consumption of about 6% of fuel consumption has to be taken into account with this concept.

As **Table 2** shows, the Euro 4 emission limits can be achieved with a 70% SCR efficiency, while due to the oxidation action of the SCR system particulate emissions are reduced by about 30%. Hence, engine-out emissions of NOx/PM of 9 / 0,04 g/kWh are required and feasible for Euro 4, as shown in the following sections.

Europe ETC, 2005 (EURO4)

Component	Legal limit [g/kWh]	Efficiency of aftertreatment system [%]	Engine raw emission [g/kWh]
PM	**0.03**	30 *)	**0.04**
NOx	**3.5**	70 (SCR)	**9.0**

*) Resulting from reduction of organic soluble particulate components by oxidative action of aftertreatment system

Table 2 EURO 4 – Emission Limits and Appertaining Raw Emissions

ENGINE CONCEPTS

Based on the required engine-out emissions as per tables 1 and 2 and the mentioned EURO 4 base engine, there are basically two routes for realizing the various low emission scenarios:

- 4 to 5 g/kWh NOx raw emissions with NOx-aftertreatment and particulate trap,
- 1 to 2 g/kWh NOx raw emissions with exhaust gas recirculation (EGR), plus NOx-aftertreatment and particulate trap.

For both routes, comprehensive studies on the basic engine by experimental work as well as by simulation were performed. The following key engine components were taken into account in establishing and evaluating various engine concepts:

- Fuel Injection System
- Exhaust Gas Recirculation
- Turbocharging
- Cooling System
- Engine Control System

CONCEPTS FOR 4 TO 5 G/KWH NOx RAW EMISSIONS

For this engine-out emission level, there are two variants possible, **Fig. 1**. Starting from a base engine of today featuring a maximum injection pressure of 1600 bar and a low swirl level of about 0,8 engine-out emissions of about 4 to 5 g/kWh NOx can be achieved in the test cycle mainly by retarded injection without applying EGR. By appropriate tuning of the combustion system about 0.07 g/kWh particulates in the ETC-test are feasible. Provided that the SCR-system achieves 80-85% NOx reduction efficiency and the DPF a 90% filtration, the emissions goal of NOx/PM of 1,0/0,01 g/kWh is achievable without essential changes of the base engine of today, Fig. 1 upper variant.

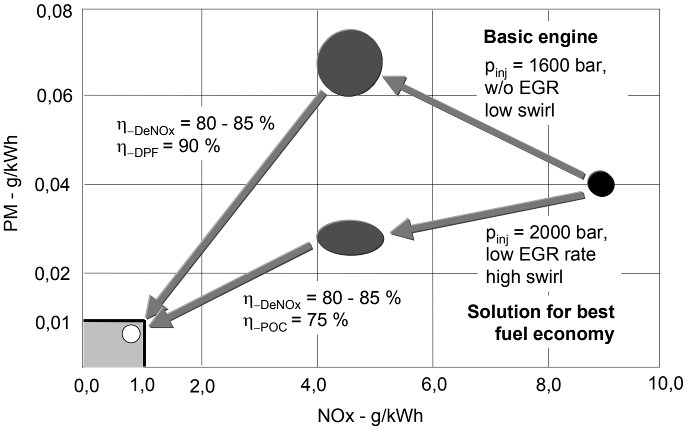


Fig. 1 Concepts for NOx = 4 - 5 g/kWh Raw Emission and Legal Limit 1,0 g/kWh

The other interesting variant in Fig. 1 takes advantage of a fuel injection system with 2000 bar pressure capability, which in combination with modest EGR rates and a higher intake swirl results in a NOx level of again 4 to 5 g/kWh, but with significantly lower PM of about 0,03 g/kWh over the ETC-test. The main advantage of the latter concept is its better fuel consumption, because NOx reduction is achieved by EGR rather than by timing retardation. The fuel consumption advantage over the ETC-test is about 7 g/kWh. When realizing the PM engine-out level of 0.03 g/kWh with the second variant in conjunction with the same

SCR-system efficiency as used with the first variant, a much simpler PM aftertreatment system could be applied - the so- called Particulate-Oxidation-Catalyst (POC). By further development of the POC, this solution should be able to achieve PM reduction efficiencies of 70 to 75% at this low engine-out emission level, thus allowing to maintain the PM engine-out limit of 0.03 g/kWh and benefit from a number of advantages. Whether such POC systems will make their way depends mainly on the longterm behaviour of their “filtration” efficiency. In any case it appears worth while to further develop this technology and bring it to volume production.

CONCEPTS FOR 1 TO 2 G/KWH NOx RAW EMISSIONS

For this topic investigations on a 2 liter displacement single cylinder research engine were performed. **Fig. 2** shows fundamental results at a bmep level of 15 bar and an engine speed of 1.500 rpm and constant start of injection. According to Fig. 2, an EGR rate of about 25 to 30% is needed to achieve an engine-out NOx level of 1 g/kWh. As shown in the lower part of Fig. 2, highest requirements to the fuel injection system are necessary in order to achieve at least 2000 bar injection pressure for soot emissions below 0.05 g/kWh. A further increase of injection pressure to 2500 bar allowes to cut the soot emission in half. These results impressively indicate the further development direction of using EGR with very high rates, if it will be followed at all.

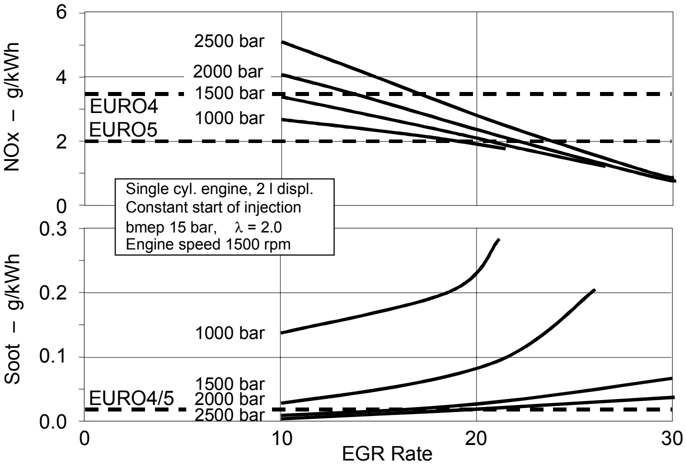


Fig. 2 Relationship between EGR Rate, Injection Pressure and NOx / Soot Emissions

These fundamental investigations lead to the development of three combustion system variants for low NOx and soot, **Fig. 3**. The basic variant 1 being suitable to achieve Euro 4 emissions is featured by 1800 bar injection pressure, about 15% EGR rate and modest intake swirl. With an injection pressure increased to 2000 bar and an EGR rate of 20%, emissions of NOx of 2 g/kWh and soot below 0.02 g/kWh could be realized according to variant 2 in Fig. 3. By further increasing the injection pressure up to 2500 bar and applying an EGR rate of about 25% at full load, and by fine tuning the whole combustion system even the extremely low engine-out emission of 1 g/kWh NOx combined with 0,02 g/kWh soot could be achieved, variant 3 in Fig. 3.

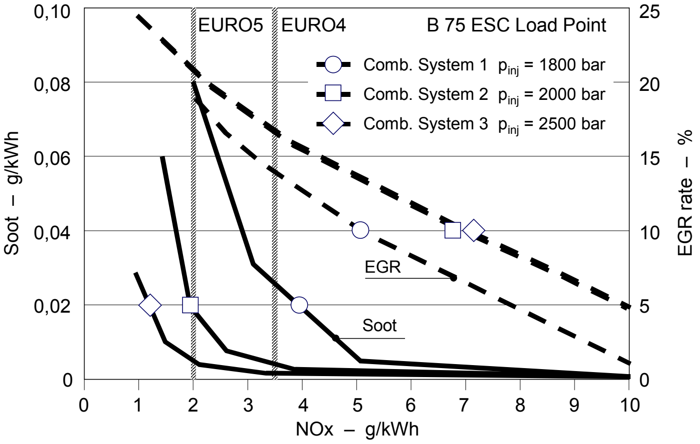


Fig. 3 Latest Development Results for Lowest Soot and NOx Engine-Out Emissions

In future emission regulations there is not only the need to meet emission standards in the various test cycles (US HDTC, ETC,...) but also within so called "not-to-exceed" areas. That means that in the whole engine map injection pressures and EGR rates as shown in **Figs. 4 and 5** will have to be maintained. In realizing the very high injection pressures special attention has to be drawn to associated combustion noise, particularly so in the part load area and during transients.

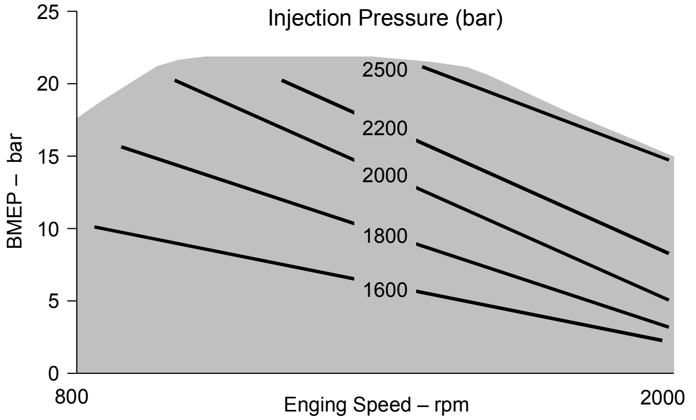


Fig. 4 Injection Pressure Demand for 1,0 g/kWh NOx and Min. Soot (engine-out)

For the NOx = 1 g/kWh engine-out concept the full load EGR rate is 25%, whereas at part load more than 40% is required. These high EGR rates result per se in such an extended ignition delay that fuel injection has ended before combustion starts. The resulting highly premixed combustion regime leads to very low NOx and nearly no soot in the exhaust. There is a twofold advantage of this behaviour, especially in conjunction with exhaust gas aftertreatment systems: by means of this "alternative combustion" higher exhaust temperatures are produced as a result of its retarded injection and combustion timing, and raw emissions of NOx and soot are minimized just in those map areas where aftertreatment systems become rather inefficient due to too low exhaust temperatures.

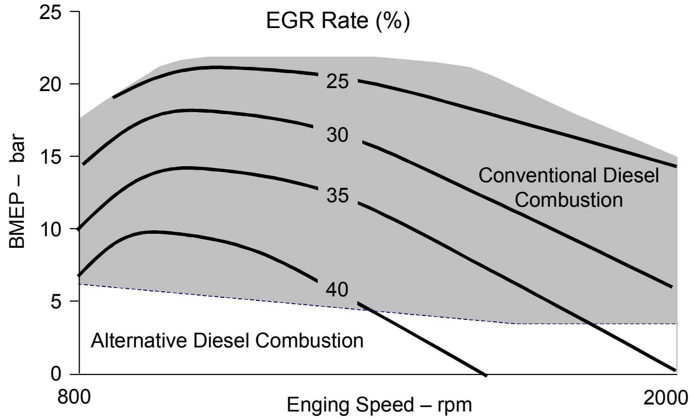


Fig. 5 EGR-Rate Demand for 1,0 g/kWh NOx and Min. Soot (engine-out)

The high EGR rates at high and full load impose highest demands on the charging system, the EGR-cooler, and on the whole cooling system of the engine. **Fig. 6** shows for the full load point bmep = 18 bar / 1.800 rpm the necessary pressure ratio of the turbo-compressor versus the EGR rate and the resulting intake manifold mixing temperature. Depending on the turbocharger overall efficiency - 45 / 50 / 55% - for a 25% EGR rate pressure ratios of 4 to 5 are required. These high pressure ratios are not only primarily the result of the high EGR rates (under the boundary condition of constant air excess ratio of about 1,5 for clean combustion) but also secondarily due to the high mixing temperatures of fresh intercooled air and cooled recirculated exhaust gas. Thus, even assuming constant EGR cooler efficiency over the engine`s life time, intake temperatures of more than 90°C are to be considered.

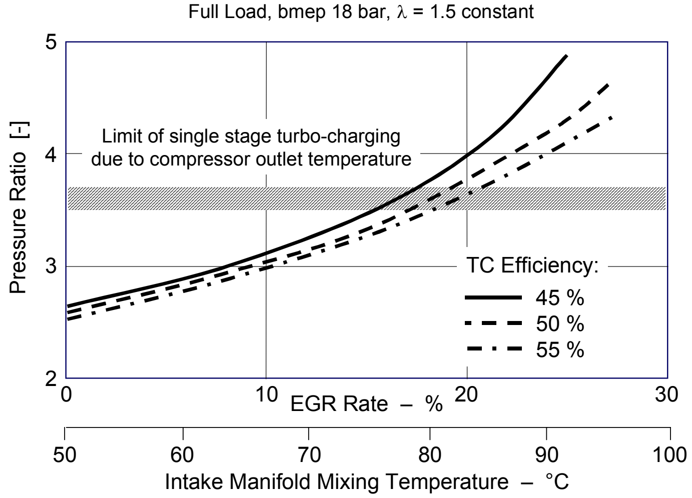


Fig. 6 Boost Pressure Demand Depending on EGR Rate and Turbo-Charger Efficiency

Taking into account that today´s compressor wheel outlet temperatures are limited at about 200 to 220°C, it becomes obvious that for achieving future pressure ratio demands two-stage turbocharging with intercooling between the first and second stage will be mandatory. In other words, given a pressure ratio limit range of 3,5 to 3.7 for single stage turbocharging, 2-stage turbocharging will be required beyond full load EGR rates of about 18 to 20%, Fig. 6.

The high boost pressure demand does not only affect the charging concept itself but leads also to higher fuel consumption due to increased pumping work as indicated by the bsfc curves in **Fig. 7** for the three different turbocharger efficiencies. The higher compressor work can only be covered by higher exhaust scavenging work which via the overall efficiency of the turbocharger turns up as additional fuel consumption. Furthermore, it has to be kept in mind that the bsfc increase in Fig. 7 is also affected by the lower combustion efficiency due to a slowed down heat release with increasing EGR rates. As a consequence taking a 25% EGR rate and today’s turbocharger overall efficiency of about 45%, the bsfc increase will be unacceptably high. This underlines the need for better turbocharger efficiencies in order to reduce fuel consumption penalties.

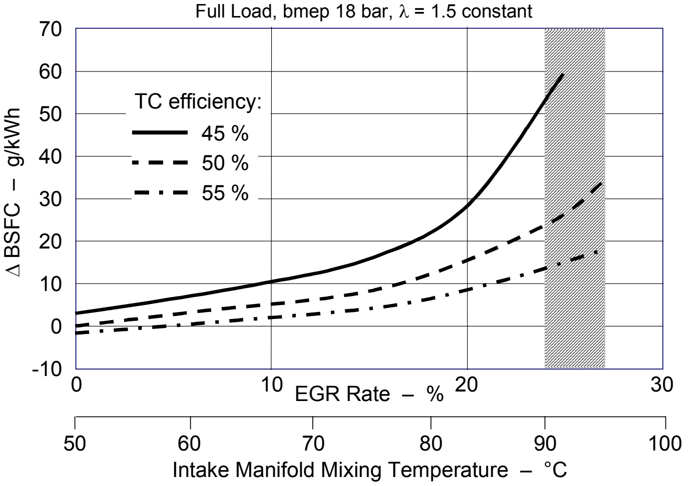


Fig. 7 Effect of EGR Rate and Turbo-Charger Efficiency on Fuel Consumption

The impact of high EGR rates and high pressure ratios on the cooling system is exemplified in **Fig. 8**, showing the amount of heat rejected from the EGR cooler, the intercoolers, and the total heat rejected into the coolant of the engine as a function of EGR rate for a 350 kW 12 liter HD diesel engine.

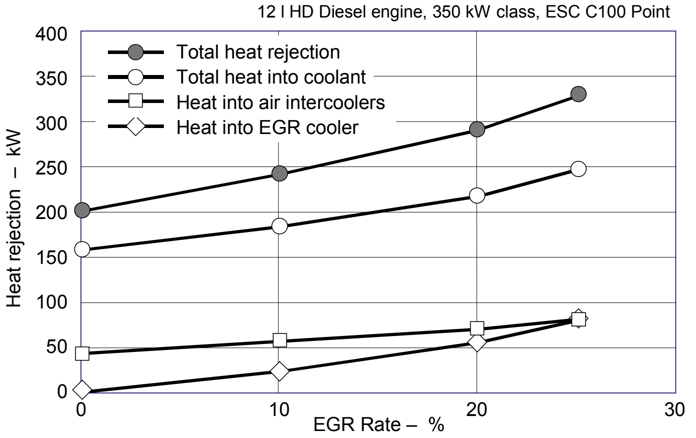


Fig. 8 Effect of EGR Rate on Heat Rejection

While today a 350 kW class engine (w/o EGR) will reject about 160 kW to the coolant at full load, with 25% EGR for 1,0 g/kWh NOx engine-out emission the heat rejection will increase to about 240 kW, due to the additional heat of about 80 kW from the EGR cooler. At the same time the heat to the charge air intercoolers increases from about 40 to 80 kW. Thus, the increase of the total cooling demand of about 130 kW, including EGR and charge air intercooling, means approximately a doubling of the required fan power.

Keeping in mind all the extreme requirements on an engine concept for NOx = 1 g/kWh engine-out emissions as discussed above, there are essentially two basic engine strategies to realise future emissions standards of US/EPA 2010 or equivalent legislations, **Fig. 9**. The first strategy is based on the assumption that over the engine's lifetime the DeNOx efficiency of the aftertreatment system is at best 80%. In this case the base engine needs to be a 1 g/kWh NOx raw emission engine. However, if the DeNOx efficiency can be maintained at 90% (as currently achieved by NOx-Adsorbers in lab tests for short operating periods only) for several 100 thousand kilometers or miles then a 2 g/kWh NOx raw emission engine would be sufficient. This second engine strategy would need "only" 2000 bar injection pressure and 20% EGR rate at full load, the latter most likely still being feasible with single stage turbocharging with increased turbocharger efficiency. Thus, promoting the further development of DeNOx systems and the improvement of turbocharger efficiencies is of high importance and could prove to be profitable for both, component suppliers as well as engine manufacturers.

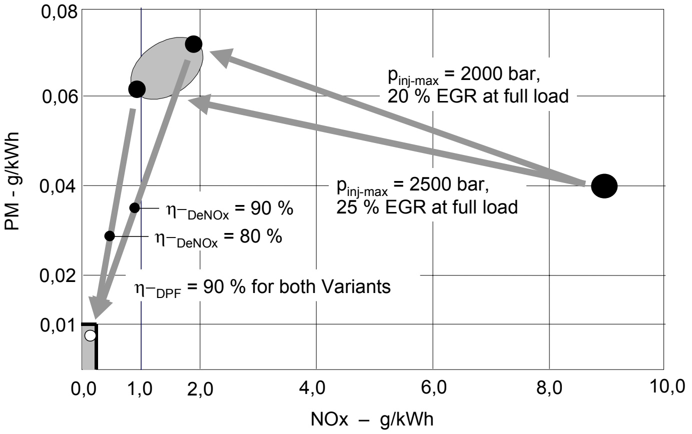


Fig. 9 Concepts for NOx = 1 - 2 g/kWh Raw Emission and Legal Limit 0,27 g/kWh

As shown so far, the crucial point is to achieve highest possible and stable conversion rates of DeNOx systems. As such also EGR systems need special attention with regard to their control quality, especially in conjunction with variable turbine geometry turbochargers (VGT). As shown in **Fig. 10** for a full load point, there is a linear decrease of specific NOx emissions with increasing EGR rate, as could be found by extensive engine testing at realistic intake mixing temperatures and EGR rates of up to 30%. The other curve in Fig. 10 shows the relative NOx change per % EGR rate. There is a significantly increasing impact of EGR control quality on NOx emissions once low NOx levels by high EGR rates are achieved. For instance at an EGR rate of 25% only slight deviations of the EGR rate result in significant differences in the specific NOx emission. As a consequence, there is a high demand on EGR control quality which so far has not been fully resolved yet.

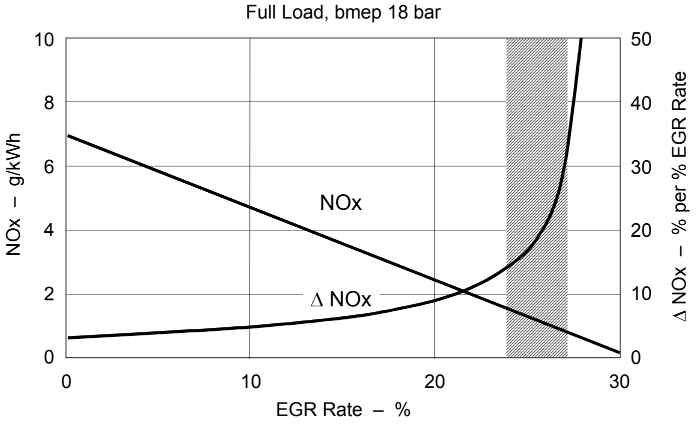
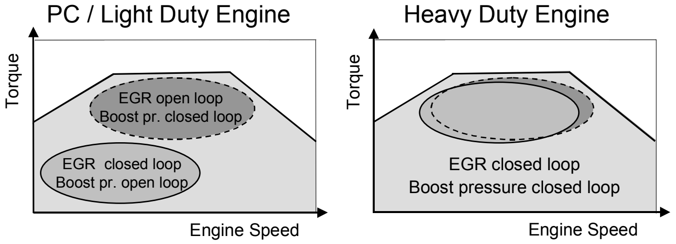


Fig. 10 Absolute and Relative Effect of EGR Rate on NOx Emissions

New ways for engine control concepts are required to overcome the basically inevitable mutual influence of boost pressure and EGR rate on each other. AVL is concentrating on model based control systems and most recently has achieved a very promising development status, as outlined in the following section.

REQUIREMENTS ON EGR RATE CONTROL

In combination with variable turbine geometry, EGR rate control requirements depend strongly on the vehicle category, **Fig. 11**. With passenger car (PC) diesel engines control areas for boost pressure and EGR rate within the engine map are relatively distant from each other due to the typical part load operating regimes in real life.



Operating Area:
- EGR and boost pressure control demand in different map areas

Operating Area:
- EGR and boost pressure control demand in the same map area

Challenge:
- Necessity of simultaneous control of EGR rate and boost pressure

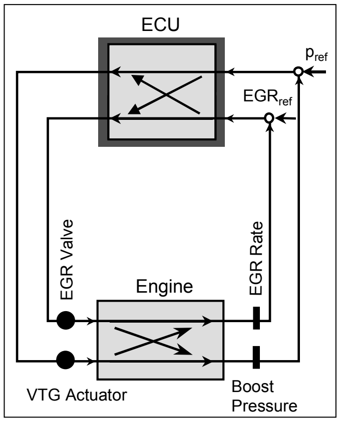
Fig. 11 EGR Rate Control Areas within Engine Map of Passenger Car and HD Diesel Engines

Thus the strategy "EGR open loop / boost pressure closed loop" at high loads and "EGR closed loop / boost pressure open loop" in the part load area is quite sufficient and already in use today. However, for HD diesel engines being operated in reality much more frequently at high loads, the two separate control areas seen with PC engines are overlaping, and the consistency of EGR rates attains first priority due to the high sensitivity of the engine in this operating area.

A solution has to be found to overcome the mutual influence of EGR rate and boost pressure on each other.

This is possible by means of model based control, **Fig. 12**, allowing the decoupling of VTG control and EGR rate control. The scheme in Fig. 12 is shown for EGR control plus VTG control, but very similar requirements of EGR control plus 2-stage turbocharging control can be covered by this concept.

By means of the model based controller it is possible to ensure the optimum combination of air and EGR rate not only during steady state but also during transient conditions.



- A **single** model based controller for EGR rate and boost pressure
- Reciprocal influence of EGR rate and boost pressure being taken into account in real engine operation

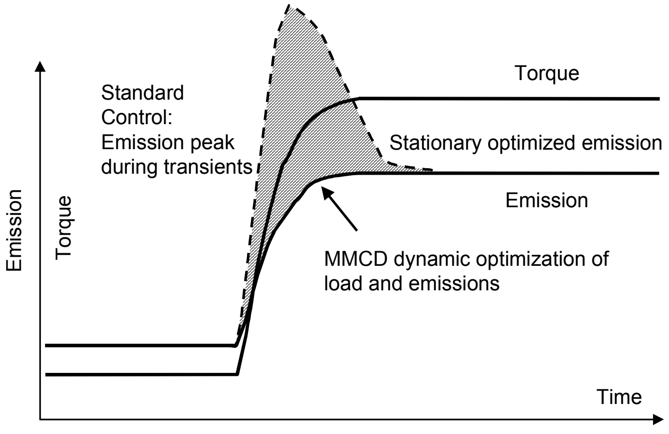
Result:

- Better dynamic engine response
- Control of EGR rate and boost pressure decoupled to a high extent
- Reduced ECU storage demand and application effort

- Emission reduction
- Fuel consumption improvement

Fig. 12 Concept of Model Based Controller for EGR and VTG

As a result, during load response / acceleration phases NOx and smoke peaks are prevented simultaneously by the model-based multi-variable controller, **Figs. 13 and 14**.



MMCD: Model-Based Multivariable Controller Design

Fig. 13 Emission Improvement by Model Based Controller during Transients

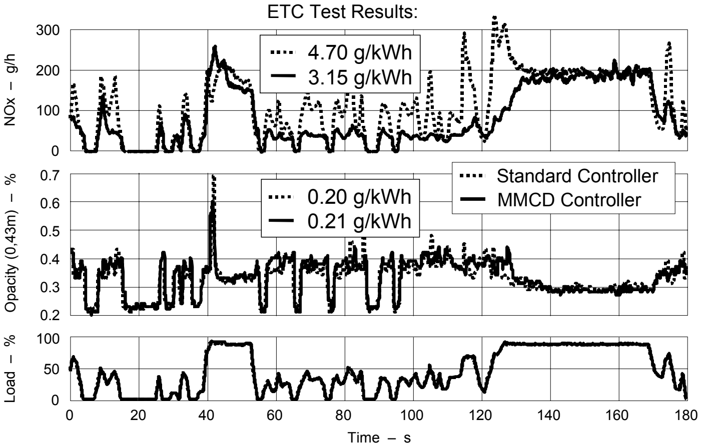


Fig. 14 Transient Emission Reduction by AVL MMCD Controller

CONCLUSIONS AND RECOMMENDATIONS

As shown by **Table 3**, different engine technologies and exhaust aftertreatment systems are required and possible, depending on the emission standards to be met. Their impact on specific fuel consumption and even more on real road fuel consumption is significant taking into account also the demand of cooling. In Table 3 road fuel consumption is set to100% for the Euro 4 engine with SCR technology (first row in Table 3) taking into account the consumption of urea-water solution at a price of 50% of diesel fuel. For the sake of better comparability of the scenarios all specific fuel consumption figures in Table 3 are based on the ETC-test.

Emission Standard NOx / PM	Raw emission NOx / PM	EGR rate [%]	Exh. after-treat-ment	Boost press. ratio [-]	PFP [bar]	Bsfc ETC [g/kWh]	Road fuel cons. (incl. urea* & fan)
3.5 / 0.03	**9.0 / 0.04**	–	SCR	2.6	200	200	100 %
1.0 / 0.01	**5.0 / 0.08**	–	SCR + DPF	2.7	180	215	107 %
	5.0 / 0.03	10	SCR + POC	3.1	200	208	105 %
0.27 / 0.01	**1.0 / 0.06**	25	SCR + DPF	4.3	200	228	115 %
	2.0 / 0.07	20	Ads.Cat? + DPF	4.0	200	218	111 %

* urea consumption calculated with 50% of diesel price

Table 3 Emission Scenarios, Engine Technologies and Consequences

Looking at the US scenario with emission limits of 0,27 g/kWh NOx and 0,01 g/kWh PM, its technology options, and resulting fuel economy deterioration, it appears promising to promote the further development of the NOx-Adsorber technology due to its basic potential of higher conversion efficiency, which in turn allows higher raw NOx emissions at better fuel consumption. With improved turbocharger efficiencies there may be even a way to avoid the application of two-stage turbocharging. Engines have to be designed for a peak firing pressure capability of 200 bar.

In case of the NOx-Adsorber the use of a fuel injection system of “only” 2000 bar could become a cost advantage.

Certainly the most difficult case is seen in Japan, if the introduction time of 2008 currently being discussed and the level of emission standards similar to those in US for 2010 will be maintained. It is to be questioned whether for this scenario the required aftertreatment technology will be available in time for mass production also taking into account the rather low exhaust gas temperature level under Japanese traffic conditions.

For Europe a scenario of NOx = 1 g/kWh and PM = 0.01 g/kWh shows interesting results. Although a significant further reduction of emissions from Euro 5 level would be achieved fuel consumption deterioration and related CO_2 emissions are kept within acceptable limits. Major engine changes relative to Euro 5 engines will most likely not be necessary at least in case the engine is not intended to be used in all world markets.

All emission scenarios for and beyond 2010, being promulgated or discussed, require further intensive development of exhaust aftertreatment systems and utmost reduction of engine-out emissions.

To meet already promulgated US/EPA 2010 emission standards requires PM filtration and DeNOx efficiencies of at least 90% in order to keep efforts needed for the base engine and fuel consumption deterioration within acceptable limits.

Emission standards discussed in Japan`s PNLTR are considered extremely critical with the introduction date 2008!

For EURO 6, still to be discussed, the right balance between further emission reduction and fuel economy / CO_2-emissions has to be found. Taking over US 2010 limits would lead to an unacceptable fuel consumption deterioration.

By 2010, fuel injection system manufacturers have to be prepared for volume supply of fully flexible injection systems with 2500 bar pressure capability.

Precise control of VTG and EGR rate requires a new approach, e.g. by means of model-based solutions.

Turbocharger manufacturers are highly challenged to develop turbochargers with total efficiencies of at least 55%.

For cooling systems new approaches regarding enhanced efficiencies and compactness have to be found.

New quality control processes enabling the extreme narrowing of emission relevant production tolerances of the total system engine plus aftertreatment have to be developed and verified.